TRANSACTIONS

AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS

VOLUME 37

THIRTY-SEVENTH ANNUAL MEETING
PITTSBURGH, PA., JANUARY 26-29, 1931

THIRTY-SEVENTH SEMI-ANNUAL MEETING
SWAMPSCOTT, MASS., JUNE 23-25, 1931

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1931

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1931

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Assistant Secretary, A. F. Bowers
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Board of Governors: G. L. Larson, W. F. Noll, F. G. Weimer

TRANSACTIONS

of

AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS

No. 877

THIRTY-SEVENTH ANNUAL MEETING, 1931

A RECORD was set at the 37th Annual Meeting of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, held at the William Penn Hotel, Pittsburgh, Pa., January 26-29, 1931, when the attendance surpassed any meeting, except that in Philadelphia in 1930 which was stimulated by the interest in the first Heating and Ventilating Exposition.

The Pittsburgh meeting was outstanding not only with its attendance of 620, but also because of the number of papers and reports presented, and the unique features of entertainment.

The technical program was particularly noteworthy, not only because of the quality of the papers presented, but also because of the variety of subjects covered and the unusually large number of papers given. Twenty-two papers were presented and in addition two codes, one dealing with concealed radiation and the other with oil burning devices, were discussed.

The 37th Annual Meeting was called to order by Pres. L. A. Harding, and F. C. Houghten, president of the Pittsburgh Chapter, delivered the address of welcome to the members and guests attending the meeting.

The Report of the Council was read by Secretary A. V. Hutchinson, and was adopted as read.

Report of the Council

The 1930 Council was organized in Philadelphia, Pa., on January 31 and during the past year has held seven meetings—two in New York City and one each in Philadelphia, Toronto, Minneapolis, Buffalo and Pittsburgh.

In compliance with the provisions of the Constitution and By-Laws, the Council appointed its standing committees: *Executive*, W. H. Carrier, Chairman; *Membership*, J. D. Cassell, Chairman; *Publication*, G. L. Larson, Chairman; and *Finance*,

F. C. McIntosh, Chairman, and during the year, as occasion demanded, appointed special and continuing committees as follows: *Guide Publication Committee,* D. S. Boyden, Chairman; *Committee on Increase of Membership,* C. W. Farrar, Chairman; *Committee on Rules of Award for F. Paul Anderson Medal,* F. C. McIntosh, Chairman; *Committee on Chapter Relations,* R. H. Carpenter, Chairman; *Committee on Revision of Constitution,* Thornton Lewis, Chairman; *Committee on Definition of Heating Terms,* A. C. Willard, Chairman; *Committee on Code for Testing and Rating Gravity Type Concealed Radiation,* R. N. Trane, Chairman; *Committee on Testing and Rating Unit Ventilators,* John Howatt, Chairman; *Committee on Testing and Rating Unit Heaters,* D. E. French, Chairman; *Committee on Revision of Code of Minimum Requirements for Heating and Ventilation of Buildings,* Perry West, Chairman; *Committee on Air Conditioning Standards,* Dr. E. V. Hill, Chairman.

At the April 2 meeting the budget of the Finance Committee was approved, which provided for an income from SOCIETY activities of $49,100 and an expenditure of $40,550 and $6,000 additional for an executive secretary of the Committee on Research, also a Guide Budget of $48,500 income and $39,900 expenditure. Both exceeded expectations. Depositories for SOCIETY funds were selected and the Endowment Fund of the SOCIETY was placed in a custodian account at the Bankers Trust Co., and the interest from this fund is to be used for research.

Dates for the Semi-Annual Meeting 1931, June 23-26, were selected and at the invitation of the Massachusetts Chapter, the meeting will be held in Swampscott. The 38th Annual Meeting is to be held January 25-29, 1932, in Cleveland, Ohio, under the direction of the Cleveland Chapter members and at the same time as the Heating and Ventilating Exposition.

Petition for a charter for a Southern California Chapter was received and acted upon favorably and the new chapter made an auspicious start with a membership of 60.

At a joint meeting with the Committee on Research the question of establishing the SOCIETY'S Research Laboratory in Buffalo in buildings offered to the SOCIETY was discussed, but it was found inadvisable to move from the present location in the Bureau of Mines Pittsburgh Experiment Station.

The Council approved the *American Standards Association* Safety Code on Mechanical Refrigeration and authorized the printing and distribution, to members for discussion, of the newly prepared Codes for Testing Steam Heating Boilers Burning Oil Fuel and the Testing and Rating of Concealed Gravity Type Radiation.

The presentation of a complete set of Society Transactions to the Bureau of Mines, Pittsburgh Experiment Station, was authorized and a portrait of Benjamin Franklin, the SOCIETY'S patron saint, was purchased for the headquarters office.

At the December meeting, the Council nominated candidates for the Research Committee as required by the Constitution and By-Laws.

Various routine actions were taken in reference to membership with the acceptance of resignations, cancellation of membership for non-payment of dues, and reinstatements. Life Membership was conferred upon the following men during the year: Henry B. Gombers, George F. Stone and Prof. L. P. Breckenridge.

At the concluding meeting of the Council on January 26, approval of a plan to have joint sessions with the *American Society of Refrigerating Engineers* at Cleveland was voted.

Respectfully submitted,

THE COUNCIL

Report of the Secretary

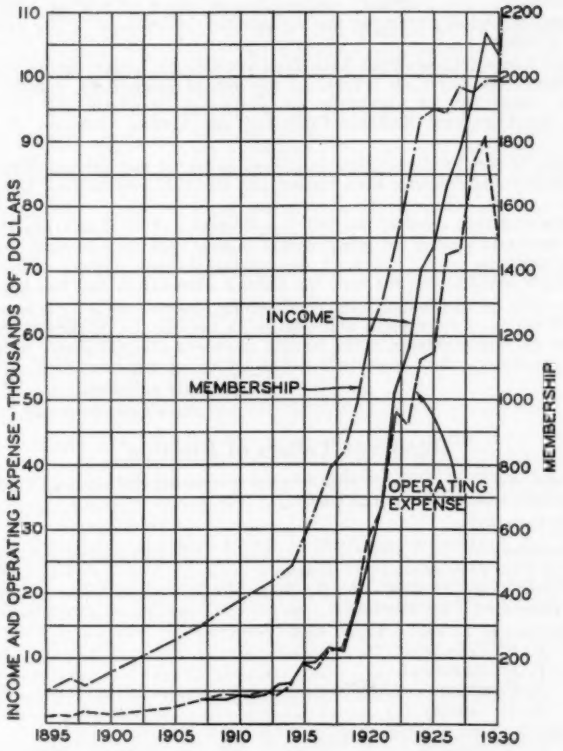
Several significant points are to be noted in a review of the Society activities for 1930: *First,* the A. S. H. V. E. Research Laboratory has successfully completed 10 years of service to the profession and industry and it is fitting that the Society should meet in Pittsburgh and hold a technical session of the Annual Meeting at the Laboratory in the U. S. Bureau of Mines Building.

Second, THE GUIDE commences its 10th year of existence and the next volume, the

1932 edition, will be the anniversary number of this important book which has made heating and ventilating history.

Third, the first year having the Society's JOURNAL as a part of *Heating, Piping and Air Conditioning,* was completed in May, 1930, with evident satisfaction to a majority of the membership.

In a glance backward over the progress of the Society since its organization in



1895, the chart showing income, expenditures and membership status, will be of particular interest as it reveals a steady growth of the organization.

During the year 1930, 354 new members were added to the roll of membership and the total number now on the roster is 2,224. It will be noticed that the Society income has increased in greater proportion than the membership, due to the large income from the JOURNAL and GUIDE.

An opportunity for rendering an extra service to the membership came this year and continued effort has been made to arrange for employment for those who needed such service.

A new location was found for the headquarters office in the New York Life Insurance Building when it was found necessary to leave the Engineering Societies' Building

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and every member who has visited the new offices has found much to praise in the action of the Executive Committee in choosing the location and furnishing the new quarters.

A closer relationship with the Chapters was established by having the National Officers give one program at each Chapter during the season. President Harding, Treasurer Farrar, Director Houghten, of the Research Laboratory, and the Secretary of the Society, visited the majority of Chapters together, and the talks given covered matters of general and technical interest as well as a comprehensive statement on the Society activities. The Secretary enjoyed the privilege of visiting each of the 15 Chapters and in October attended the organization ceremonies of the new Southern California Chapter.

All of the Council Committees have been exceedingly active during the year and heavy burdens have fallen on several of the special committees, notably the Guide Publication Committee, the Committee on Ventilation Standards and the committees developing the Concealed Radiator Code and the Testing Code for Boilers using Oil Fuel.

Two copies of the TRANSACTIONS have been prepared and printed during the past year, the 1928 volume having been mailed and the 1929 edition now being prepared for mailing.

All routine business required by the Constitution and By-Laws has been carried out by the Secretary's office and the activities during 1930 have been of larger volume than in any previous year. The work was carried through without additional assistance and much credit is due the staff for the fine cooperation that has been given.

The year 1931 is anticipated with considerable pleasure as it is believed that the Society will maintain its growth and add to its prestige by attracting the attention and interest of a larger number of the leading engineers who will desire to be identified with an organization which pays such a fine return on the investment.

Respectfully submitted,

A. V. HUTCHINSON, *Secretary.*

Report of Tellers of Election

The undersigned Tellers report that they have counted and checked all ballots cast and find the following to be a true count:

OFFICERS—	
President—W. H. Carrier	654
First Vice President—F. B. Rowley	653
Second Vice President—W. T. Jones	653
Treasurer—F. D. Mensing	654
Members of Council—Three Year Term:	
E. K. Campbell	655
E. O. Eastwood	655
R. Farnham	654
E. H. Gurney	655
Council—Two Year Term:	
D. S. Boyden	655
Void Ballots	3
Members of Committee on Research:	
C. A. Booth	642
F. B. Howell	641
Walter Klie	643
F. B. Rowley	643
J. H. Walker	644
Void Ballots	2
Total ballots cast not signed	11
Total ballots cast by members in arrears of dues	20

Respectfully submitted,

S. E. DIBBLE, *Chairman;*
ALFRED J. OFFNER, F. H. MORRIS.

Report of the 1931 Guide Publication Committee

THE GUIDE 1931 is now available. In order to maintain the leadership of the A. S. H. V. E. GUIDE as the accepted authority of the heating and ventilating profession, it is necessary to coordinate the results of the latest research with the practice of the outstanding engineers of the profession by recording the most up-to-date information and data available on each phase.

The editorial staff selected by the Guide Publication Committee, therefore, includes heating and ventilating engineers of unusual qualifications, who are specialists in the subjects to which they are invited to contribute.

The Guide Publication Committee is greatly indebted to the following engineers through whose efforts THE GUIDE 1931 has been made possible, and who have given their services without compensation or thought of reward other than the satisfaction of having performed a public service:

	W. G. Frank	
O. W. Armspach	S. McK. Gray	H. C. Murphy
W. S. Bondy	W. E. Heibel	Percy Nicholls
D. R. Brewster	R. H. Heilman	H. W. Pfeffer
J. C. Butler	F. C. Houghten	W. A. Rowe
A. C. Davis	H. B. Johns	G. K. Saurwein
T. J. Duffield	R. L. Jones	W. M. Sawdon
J. E. Emswiler	F. A. Kitchen	W. H. Severns
J. L. Faden	L. L. Lewis	Perry West
P. E. Fansler	E. A. May	H. M. Wylie

Total number of books printed	11,500
Total number of advertisers	203
New advertisers	46
Number of pages—Text Section	544
Number of pages—Advertising Section	360
Number of pages—Membership Section	60

The Committee has felt that the saturation point of usefulness of THE GUIDE might in the near future be reached, and in view of the present business depression the Committee was concerned as to the amount of advertising which might be secured.

The income from THE GUIDE 1931 was $52,374.31, with expenditures of $33,780.09, or a net balance of $18,594.22.

The above indicates that there is still an increasing demand for the yearly publication of THE GUIDE.

Attention is called to the fact that some manufacturers during periods of depression increase advertising in an effort to stimulate sales, which fact is further evidence that THE GUIDE may continue indefinitely as a yearly publication.

THE GUIDE PUBLICATION COMMITTEE,
D. S. BOYDEN, *Chairman;*
V. S. DAY, R. V. FROST, J. F. MCINTIRE, S. R. LEWIS.

J. D. Cassell reported for the Membership Committee and stated that 354 new members had been secured during the past year, which he intimated was satisfactory in view of existing conditions.

Report of Finance Committee

The past year has not been a particularly prosperous one. More men than usual were obliged to cut down all possible expenses and the collection of dues was affected. Advertisers in THE GUIDE were inclined to reduce their advertising appropriations and the financial success of this, our other main source of revenue, was threatened.

However, the year turned out to be a profitable one for us. Due to the conscientious and effective work of Mr. Farrar, the total membership was increased sufficiently to give a net gain of over $2,000 from initiation fees and dues. Our normal expenses

for the year were held to about the same as for 1929, and other sources of income, such as sales of TRANSACTIONS, reprints and books, were increased enough to pay the expenses of moving to our new offices and of procuring the additional furniture and equipment needed.

In spite of increased resistance, Mr. Hutchinson was able, by great effort, to increase the income from advertisements in THE GUIDE by 12½ per cent and to increase the copy sales by 15 per cent. The cost of the promotion work was greater than estimated, but the extra cost was more than offset by the saving in production costs. Mr. Hutchinson deserves as much credit for reducing expenses as for increasing the revenue.

As a result of these conditions, our General Fund has been increased from $28,591.56 to $32,188.72 and THE GUIDE profit available for turning over to the Research Laboratory will be more than last year. The full report of our finances and operations was made by our certified public accountant, Frank G. Tusa.

The amendment to our Constitution fostered by Mr. Jones and adopted at Philadelphia a year ago requires that our funds be invested in securities listed as legal for investment of savings bank funds in the State of New York. Because of the general depression in the bond market and the fact that the higher yield bonds we have held are naturally more depressed than the low yield securities we must buy, it has been thought inadvisable to sacrifice principal by immediate reinvestments. However, as it has been possible to sell about 30 per cent of our securities at a slight profit, we have done so and invested the proceeds according to the amendment. This policy of the Council will probably be continued so as to make reinvestments at the most favorable times.

Respectfully submitted,

F. C. MCINTOSH, *Chairman;*
THORNTON LEWIS, W. T. JONES.

The foregoing report was duly approved by vote.

Report of Certified Public Accountant

January 15, 1931.

AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS,
51 MADISON AVENUE
NEW YORK CITY

Gentlemen:

Pursuant to your request I made an examination of the books of account and records of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, New York City, for the year ending December 31, 1930, and submit herewith my report.

The work covered a verification of the Assets and Liabilities as of the date previously stated and a review of the operating accounts. For the past calendar year, the recorded cash receipts were traced into the depositories; the cancelled bank vouchers were compared with the cash disbursement records; and the membership dues were accounted for.

Submitted herewith is a Balance Sheet showing the financial condition of the Society on December 31, 1930, and your attention is directed to the following comments thereon:

CASH

Cash on deposit was verified by direct communication with the banks listed below and by reconcilement of the amounts reported to me with the balances shown by the books of the Society:

BANKS	AMOUNTS
Chase National Bank (Special) Treasurer	$2,283.21
Chase National Bank (Regular) Secretary	1,775.04
Bankers Trust Company (Special Interest)	1,425.25
Bank of America N. A.	113.70
Bank of United States	500.00
Federation Bank and Trust Company (Pass Book No. 28992)	2,628.55
	$8,725.75

The petty cash on hand was counted and found correct.

MARKETABLE SECURITIES

There is attached hereto a schedule of negotiable bonds which were verified by me either by direct communication with the Bankers Trust Company where same are deposited for safekeeping or by count of those on hand. No adjustment has been made of the $5,805.67 shrinkage in the market value of these securities, instead I have included them on the Balance Sheet at cost and indicated the market value of December 31, 1930 adjacent thereto.

ACCOUNTS RECEIVABLE

Unpaid membership dues were determined by trial balance of the individual ledger cards and comparison of same with the Dues Register. A summary of these as to years charged is shown below:

1930 Unpaid Dues	$9,889.24
1929 Unpaid Dues	3,382.65
1928 Unpaid Dues	560.00
Total	$13,831.89

My verification of the dues disclosed that during the year 1930 there had been prepaid to the Society dues amounting to $509.65. These I have shown on the attached Balance Sheet as deferred income.

Based upon previous experience it is estimated that only one-third of the current year's dues are realizable. Accordingly I have provided a Reserve for 1930 Dues Doubtful of Collection of $6,592.83. This is equivalent to two-thirds of the 1930 Unpaid Dues. After reviewing the 1928 and 1929 Dues it appears that very little will be realized from this source, therefore I have increased the reserve for prior years' dues doubtful of collection to $3,643.08.

A comparison of the membership in force as of the close of business, December 31, 1929 and 1930, respectively, follows:

CLASSIFICATION	1930	1929	INCREASES
Members	1,485	1,424	61
Associates	480	405	75
Juniors	246	156	90
Students	16	1	15
Honorary	2	2	0
	2,229	1,988	241

Trial Balances were also taken of Accounts Receivable resulting from Advertisements and sales of Guides, Codes, Books and Reprints. All available data examined and a discussion with the management relative to these accounts shows that the reserve of $1,000.00 provided on the books is sufficient to cover future contingencies.

INVENTORIES

There is scheduled below the inventory of Transactions taken on December 31, 1930:

YEAR	VOLUME	NUMBER	PRICE	AMOUNT
1923	29	142	$1.97½	$280.45
1924	30	280	1.38¼	387.11
1925	31	174	.96⅛	167.34
1926	32	114	1.26⅛	144.57
1927	33	163	1.32⅛	316.45
1928	34	232	1.43	331.76
		Total		$1,627.68

Transactions covering the year 1929, Volume 35, and the year 1930, Volume 36, were not published up to the close of the past year, therefore reserves of $3,000.00 each have been provided to cover the future cost of producing the aforementioned volumes.

All other Inventories were verified either by actual count or analysis of the records.

ACCOUNTS PAYABLE

A list was compiled of all invoices remaining unpaid on December 31, 1930 dating prior to January 1, 1931, for the purpose of determining all Accounts Payable and a liability therefore was set up in the sum of $15,169.77. This sum includes charges

from Horn-Shafer Company of $13,270.00 for costs covering the printing of THE GUIDE AND YEAR BOOK.

DUE RESEARCH LABORATORY

Of the dues charged to Members and Associates, 40 per cent has been reserved for the Research Laboratory in accordance with Section 5, Article 3 of the By-Laws. The sum payable to the Research Laboratory as and when the Dues Receivable will have been realizd in cash is $16,788.10.

ACCRUED ACCOUNTS

A Bonus of $7,809.57 to the staff of the Society has been included as a liability on the subjoined Balance Sheet. The computation of this Bonus has been made in accordance with a resolution adopted by your council at a meeting held on June 24, 1930.

GENERAL FUND

An Analysis of the General Fund of the Society showing the changes made therein during the year under audit follows:

General Fund December 31, 1929—per Former Report		$28,591.56
Additions:		
Profit from Society Activities for the Calendar Year 1930 from Society Statement of Income and Expenses	$3,538.94	
Add: Amount assumed by Society for F. Paul Anderson Medal Award included as an expense in the Statement previously mentioned	500.00	
	$4,038.94	
Profit from Guide for the Calendar Year 1930 from Guide Statement of Income and Expenses	9,784.65	
Adjustment of items covering Council and Special Meetings expenses assumed by Society in 1929 which are chargeable to Research	311.38	14,134.97
		$42,726.53
Deductions:		
Additional amount set aside for Endowment Fund to bring said fund to the Cost Value of $20,000.00 par value of Bonds voted by Council at a meeting held on June 24, 1930	9,337.81	
Increase in Reserve for Prior Years' Dues Doubtful of Collection	1,200.00	10,537.81
General Fund December 31, 1930, per Balance Sheet		$32,188.72

Respectfully submitted,

FRANK G. TUSA,
Certified Public Accountant.

BALANCE SHEET

AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS
December 31, 1930.

ASSETS

SOCIETY			
CASH			
On Deposit		$ 8,725.75	
On Hand		100.00	$ 8,825.75
INVESTMENTS (AT COST)			
Securities (Market Value $12,760.00)		12,919.49	
ADD: Accrued Interest		152.92	13,072.41
ACCOUNTS RECEIVABLE			
Membership Dues	$13,831.89		
LESS: Reserve for Doubtful	10,235.91	3,595.98	
Advertisements	43,582.68		
Guides	1,284.25		

Codes	$108.55			
Books and Reprints	102.71			
	45,078.19			
LESS: Reserve for Doubtful	1,000.00	$44,078.19	$47,674.17	
INVENTORIES				
Transactions 1923-1928		1,627.68		
Transactions 1929 in Process		513.46		
Transactions 1930 in Process		50.86		
Emblems		86.35		
Postage—General		277.40	2,555.75	
PERMANENT				
Library		300.00		
Furniture and Fixtures	7,451.39			
LESS: Reserve for Depreciation	3,985.68	3,465.71	3,765.71	$75,893.79
SPECIFIC FUNDS				
ENDOWMENT FUND				
Securities at Cost (Market Value $14,350.00)		19,753.68		
ADD: Accrued Interest		250.83	20,004.51	
F. PAUL ANDERSON AWARD FUND				
Cash on Deposit			1,023.43	
RESEARCH LABORATORY FUND				
Cash on Deposit—Research	5,365.60			
Cash on Deposit—Research Endowment Fund	610.04			
Cash on Hand for Deposit	21.00	5,996.64		
Securities at Cost (Market Value $3,015.00)	3,007.50			
ADD: Accrued Interest	45.00	3,052.50	9,049.14	30,077.08
				$105,970.87

LIABILITIES AND CAPITAL

SOCIETY			
ACCOUNTS PAYABLE		$15,169.77	
DUE RESEARCH LABORATORY			
Dues	$13,191.08		
Interest from Endowment Funds	1,025.00	14,216.08	
ACCRUED ACCOUNTS			
Bonus to Staff		7,809.57	
RESERVE FOR TRANSACTIONS			
1929	3,000.00		
1930	3,000.00	6,000.00	
DEFERRED INCOME			
Prepaid Dues		509.65	
TOTAL LIABILITIES		43,705.07	
GENERAL FUND			
Society		32,188.72	$75,893.79

SPECIFIC FUNDS

Endowment		$20,004.51	
F. Paul Anderson Award		1,023.43	
Research Laboratories	$8,439.10		
Research Endowment Fund	610.04	9,049.14	$30,077.08
			$105,970.87

Note "A": The General Fund of the Society is subject to future adjustment pending action to be taken by Council on the assignment of the Guide Profit for the Calendar Year 1930.

Note "B": This Balance Sheet is subject to the comments contained in the letter attached to and forming a part of this report.

The Report of the Committee on Code for Testing and Rating Concealed Gravity Type Radiation was submitted by R. N. Trane, chairman, who explained in detail the principles of the code and the ideas of his committee, consisting of E. H. Beling, W. F. Goodnow, Hugo Hutzel, A. P. Kratz, E. J. Vermere and O. G. Wendel. A preliminary draft of this code was sent to members of the Society to be used as a basis of discussion. Mr. Trane related some of the experiences of the committee in developing the code, and stated that this preliminary draft could not be considered final, but represented what was believed to be the best available information.

After a lively parliamentary discussion in reference to making the code available for use, by Messrs. French, Mensing, Fleisher, Rowe, McIntire and Holton, the following resolution was offered by Mr. Rowe, seconded by Mr. McIntire, and adopted:

Resolved that, the Society recommend the use of the steam section of the Report of the Committee on the Proposed Code for Testing and Rating Concealed Gravity Radiation, and that the suggestions developed in this meeting be referred to the committee and incorporated in its report at the next meeting of the Society.

The report of the Committee on Oil-Burning Devices was presented by R. V. Frost, in the absence of L. E. Seeley, Chairman. The report consisted of the presentation of the proposed code for Testing Steam Heating Boilers Burning Oil Fuel, copies of which had previously been sent to members of the Society.

The Technical Advisory Committee on Oil Burning Devices, which prepared this proposed code, consisted of L. E. Seeley, *Chairman,* H. F. Tapp, P. E. Fansler, H. R. Linn, J. H. McIlvaine, R. V. Frost and F. C. Houghten. The major divisions of the code consist of Definitions, Set-up of Oil Burner and Boiler to be Tested, Instruments and Measuring Apparatus, Fuel Sampling, Duration of Tests, Starting and Stopping Continuous Tests, Starting and Stopping of Intermittent Tests, Method of Conducting Test and Performance Charts.

Written discussions by C. H. Flink and H. M. Hart were presented and suggestions and criticisms were offered by R. F. Connell, C. E. Bronson J. F. McIntire and F. W. Hvoslef.

Mr. Frost stated that the suggestions would be passed on to the committee, and he moved:

That the report of the committee be accepted and referred to a continuing committee with instructions that the criticisms submitted at the meeting be considered and embodied in the final draft of the code.

The motion was carried.

Report of the Committee on Research

By F. B. ROWLEY, *Chairman*

The work of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS' Research Laboratory covers a wide range of interests, and one of the difficulties is to present in a condensed report the results of these activities so that the members of the Society who are of necessity more or less disconnected with them throughout the year may become reasonably familiar with the work.

The work of the Laboratory is under the supervision of a committee of fifteen members and a director. The general committee has held three meetings during the year. However, due to the difficulty of calling a large committee together, the greater part of the work has been handled through the Executive Committee, the Director, and the Technical Advisory Committees for the various projects. As in the past, the policy has been to conduct as much of the work as possible at the Pittsburgh Laboratory and to maintain cooperative projects with universities and other institutions where the proper facilities and interests are available.

While the rules and regulations governing the Laboratory's activities passed at the 1930 annual meeting anticipated that the Laboratory might conduct commercial tests, only one request has been received for this type of work during the past year. This was a request from the *Heating and Piping Contractors' National Association* for the rating of radiators. Plans for procedure were prepared by the Technical Advisory Committee on Radiation, and the Association was advised of the conditions under which the Laboratory proposed to make the tests. Due to the adverse financial conditions for the year, however, the Association decided not to go ahead with the work. Due to the difficulty of finding a proper place for conducting various commercial tests under the existing arrangements for the Laboratory, it seems doubtful whether commercial testing is advisable until such time as specific arrangements can be made for laboratory facilities to take care of it.

After working for a year under the new Laboratory regulations, there are certain minor changes which it appears advisable to make, some of them to make the regulations conform with the Constitution of the Society, and others to simplify the operation of the Committee. D. S. Boyden has been appointed as a committee of one to go over the regulations and recommend the necessary changes to the Society for adoption.

Technical Advisory Committees

The various projects for the year have been under the direction of eleven Technical Advisory Committees. The work of these various committees is covered in the report of F. C. Houghten, Director.

Cooperative Research

The institutions at which cooperative research programs are carried on and the subjects under investigation at these institutions are as follows:

Armour Institute: Measurement of Air Flow Through Registers and Grilles.

Harvard School of Public Health: The Ionization of Air and Its Relation to Health.

Washington University: Atmospheric Dust and Smoke.

University of Wisconsin: Infiltration of Air and the Drift of Air Through Buildings.

University of Minnesota: Transmission of Heat Through Building Materials, a Study of Air Cleaning Devices, and The Thermal Properties of the Different Species of Wood, in co-operation with the *National Lumber Manufacturers' Association.*

Texas Agricultural and Mechanical College: Pipe Sizes for Steam and Hot Water Heating.

Carnegie Institute of Technology: Capacity of Pipes for Various Parts of a Steam Heating System.

Yale University: Study of Oil Heating Devices.

Co-operative research projects are under consideration by other institutions, as,

the University of Illinois, Purdue University, and the Massachusetts Institute of Technology.

The general plan of the co-operative projects is that the finances are furnished equally by the Research Laboratory and the co-operating institution, with space, equipment, and supervision furnished by the institution. By this arrangement, the Laboratory is able to accomplish very much greater results with a small amount of money than would otherwise be possible.

FINANCIAL REPORT OF RESEARCH LABORATORY

COMMITTEE ON RESEARCH
AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS
TOTAL INCOME AND EXPENSES 1923 TO 1930

	INCOME					
YRS.	DUES (40%)	GUIDE PROFITS	CONTRIBUTIONS	INTEREST	TOTAL RECEIPTS	EXPENDITURES
1923	.....	.....	.....	.....	$28,903.98	$33,235.43
1924	.....	.....	.....	.....	16,920.29	21,519.91
1925	$13,604.88	.....	$13,512.57	.....	27,117.45	17,623.82
1926	22,597.53	$6,994.24	5,805.00	$322.05	35,718.82	26,607.49
1927	13,513.54	4,401.60	14,209.04	289.84	32,414.02	35,324.83
1928	12,722.96	4,022.55	12,690.32	144.38	29,580.21	32,017.90
1929	16,941.47	6,227.20	10,257.87	110.05	33,536.59	33,683.15
1930	14,104.75	4,973.50	26,720.28	340.40	46,138.93	40,486.26

FINANCIAL STATEMENT
A. S. H. V. E. LABORATORY
JANUARY 1, 1931

Research Fund (Dec. 31, 1930)			
N. Y. Bank		$ 5,386.60	
Pittsburgh Bank		1,695.38	
Petty Cash		46.66	
Other funds 1. Research Endowment		610.04	
2. Securities and Interest		3,052.50	
3. Due Research from A. S. H. V. E. Dues		13,191.08	
4. Due Research from A. S. H. V. E. Endowment Fund		1,025.00	
			$25,007.26

STATEMENT 1930 OPERATIONS

INCOME			
Balance (Dec. 31, 1929)		4,763.47	
Receipts from A. S. H. V. E. Dues and Guide		19,078.25	
Receipts from Contributions and Contracts		26,720.28	
Interest from Bank acct.	$176.01		
Securities	164.39	340.40	
			50,902.40
EXPENSES			
Salaries (Director and staff)		23,310.50	
Travel (Director and staff)		1,791.01	
Travel (Committee Members)		761.32	
Lab. Equipment and Supplies		1,697.45	
Office Expense		848.28	
Meetings		103.45	
Executive Secretary		2,041.75	
University Cooperation		9,932.50	
			40,486.26

COOPERATIVE AGREEMENTS

	BUDGET	PAID 1930
Armour Institute of Technology	$800.00	$700.00
Harvard University	500.00	500.00
Texas A. and M. College	250.00	250.00
Texas Engineering Experiment Sta.	650.00	400.00
University of Minnesota	1,750.00	1,750.00
University of Minnesota (Special)	2,500.00	2,500.00
Washington University	70.00	70.00
Yale University	1,812.50	1,762.50
University of Wisconsin	1,500.00	1,500.00
Assn. for Cor. Ther. Res.	500.00	500.00
	$10,332.50	$9,932.50

One of the outstanding features of the work for the year is that, in spite of the difficult financial times, the Laboratory budget has continued to grow, which proves the interest of the members, the industry, and various associations in the work of the Society and the Laboratory. It is significant that the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS is the only national society which maintains such a laboratory, and the evident interest in the work indicates that the Research Laboratory has a very bright future.

Report of the Director

By F. C. HOUGHTEN

The year 1930 has witnessed marked progress in the various activities of the Research Laboratory of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS. The work of the Laboratory is divided into eleven phases covering separate branches of our engineering knowledge; each phase of the work is carried on either in the Laboratory at Pittsburgh or in one or more of the eight co-operating institutions under the guidance of one of the technical advisory committees referred to in this report.

Marked progress was made on all the eleven research investigations. The work on some of these activities was brought to a sufficient stage of completion to result in conclusive and acceptable data of such a nature as to be usable in THE GUIDE.

Nine technical papers for presentation at the 1930 Semi-Annual and the 1931 Annual Meetings of the Society were completed during the year.

1. Pipe Sizes for Steam and Hot Water Heating Systems.—*Technical Advisory Committee:* H. M. Hart, *Chairman;* F. E. Giesecke, R. C. Morgan, C. V. Haynes, S. E. Dibble.

a. Capacity of Pipe for Various Parts of a Hot Water Heating System.

The study of pipe sizes for hot water heating systems is carried on by Professor Giesecke and his associates at the Agricultural and Mechanical College of Texas through a co-operative agreement with the Laboratory. This work has for its primary object the development of pipe size tables for laying out hot water heating systems which may be incorporated in THE GUIDE.

Professor Giesecke has had this work under way for several years and at the 1930 Annual Meeting, Prof. E. G. Smith of the A. & M. College of Texas, presented a very comprehensive paper, which included a method for laying out the various parts of a hot water heating system. The values are being rechecked before this report is accepted as final. Additional information on the subject in the nature of a technical paper is promised for the near future.

In addition to the work on actual hot water heating systems, Professor Giesecke is also making a study of the loss in head due to the flow of water through various fittings, including ells and tees. The work particularly includes a study of the loss in head for water flowing through two branches of a tee, a subject on which little or no data are available at present.

b. The Capacity of Pipe for Various Parts of a Steam Heating System.

The study of pipe sizes for steam heating systems began in 1922 and has continued to date. This work is carried on by the Research Laboratory at the Carnegie Institute of Technology, Pittsburgh. During the past two years the work has been directed toward the determination of pipe sizes for the return side of the system. In this connection, a report was presented at the 1930 Summer Meeting of the Society, and published in the August 1930, JOURNAL. The report gave the relation between pressure drop and the flow of water and air in horizontal pipe. A paper entitled Capacity of Return Risers for Steam and Vapor Heating Systems was published in the October JOURNAL of the Society for presentation at the 1931 annual meeting. These two papers form the basis for a complete revision of the return sizes for steam and vapor heating systems as contained in THE GUIDE. The revision of the practical tables for use in THE GUIDE will be made through co-operation between the Labora-

tory and the Technical Advisory Committee and others interested in the practical application of the results. This is desirable since the development of such tables involves the question of allowing a suitable factor of safety over the capacities actually found in the Laboratory as well as such questions as the most desirable heating-up period.

The return pipe sizes in THE GUIDE tables will require a complete revision of these values based upon the Laboratory's findings since the work was undertaken in 1922. Additional work on pipe sizes which the Laboratory has under consideration includes a check of the application of the Babcock formula to the flow of steam in modern pipe. In recent years the art of making pipe has improved with the result that pipe now on the market has a smoother internal surface and contains fewer constrictions. This development has apparently resulted in a change in the flow pressure-drop relationships for the flow of steam.

2. Study of the Application of Copper Tubing to Steam and Hot Water Heating Systems.—*Technical Advisory Committee:* S. R. Lewis, *Chairman;* W. M. Robinson, J. F. McIntire, A. P. Kratz, H. M. Hart, F. E. Giesecke.

This is a new study recently inaugurated by the Laboratory in co-operation with the *Associated Copper Tubing Manufacturers,* whereby this Association is financing the entire study. The work includes a determination of the capacity of copper tubing for carrying steam in various parts of steam and hot water heating systems and also the heat emission from such tubing when bare, and when insulated with pipe covering. The study of the capacity of this tubing for steam heating systems is being made at the Laboratory in Pittsburgh. The work on the use of such tubing in hot water heating systems will be carried on by Professor Giesecke at the A. & M. College of Texas.

3. Garage Ventilation.—*Technical Advisory Committee:* E. K. Campbell, *Chairman;* E. B. Langenberg, W. C. Randall, A. C. Davis, A. R. Acheson.

The study of garage ventilation was undertaken by the Laboratory about a year ago and had as its object a study of the relation of the quantity of air handled, its temperature, velocity, and location of supply and exhaust and its quantity of the outside air added, to the concentration and distribution of carbon monoxide, gasoline vapors, and other obnoxious fumes in the garage. A few tests were made early in the year by Dean Langsdorf of Washington University in co-operation with the Laboratory in which the condition of the atmosphere in a certain garage in St. Louis was studied. The mild climate prevailing in St. Louis in the winter and the infrequency with which the windows and doors of the garage were closed, placed the investigators at a disadvantage and it was decided to remove the investigation to a colder climate. The Laboratory is now negotiating with Massachusetts Institute of Technology for a co-operative study of this subject. A committee of the Massachusetts Chapter is co-operating with the Laboratory in this connection.

4. Transmission of Heat Through Building Construction and Building Materials.—*Technical Advisory Committee:* A. E. Stacey, *Chairman;* P. D. Close, A. B. Algren, H. J. Schweim, A. P. Kratz.

a. Determination of Heat Transfer Through Built Up Wall Sections by the Guarded Hot Box.

Professor Rowley at the University of Minnesota, in co-operation with the Research Laboratory, has been continuing this study throughout the year. Many standard walls containing various combinations of the more common building materials have been checked in the guarded hot box, resulting in more acceptable coefficients for these types of construction. This study has also served to verify the application of the accepted formula to the calculation of overall transfer co-efficients from the conductivity, air space, and film conductance co-efficients of the component parts of a wall.

b. Determination of Conductivity of Homogeneous Building and Insulating Materials.

This work is carried on at the University of Minnesota in co-operation with the Research Laboratory. In order to replace the many and varied co-efficients of conductivity for various simple building materials contained in THE GUIDE, with a

consistent set of values obtained from materials now on the market and in common use, a study has been made of the conductivity of such materials as purchased on the open market. These tests are made in the guarded hot plate in accordance with the tentative code of the *National Research Council*. Conductivities have already been determined for a large number of materials purchased on the open market. In each case, three separate tests are made on samples obtained in different localities of the country. It is hoped that a technical paper, presenting the results of this study, will be available for publication in the spring and that the tabulated results may find their way into THE GUIDE in 1932.

c. Film Conductance Coefficient for Various Types of Surfaces for Still and Moving Air

Professor Rowley at the University of Minnesota, in co-operation with the Laboratory, has made a determination of film conductance coefficients for a number of surfaces for still and moving air for a condition where the direction of the wind is parallel to the surface and measured in the center of a 6-in. duct. A paper entitled Surface Conductances as Affected by Air Velocity, Temperature, and Character of Surface was published in the June 1930 JOURNAL, and presented to the Society at the 1930 summer meeting.

An allied study of another phase of the same study was made at the Laboratory in Pittsburgh. The object of this study was to establish information concerning the relation between wind velocity in free space and the existing velocities near a building wall, and also the effect of velocities as measured at various distances away from a wall on the resulting film conductance coefficient.

In this connection, the Laboratory made a study of wind velocity gradients near a wall surface. This study has so far been limited to a study of such gradients when the wind direction was parallel to the wall surface. The most of the data collected were for a wall attached to the side of a truck and moved forward through the air so as to produce the equivalent of a wind velocity, while the velocity and static pressures near the wall surface were surveyed with Pitot tubes. Similar velocity gradients were analyzed for a 6-in. and a 12-in. duct with wind blown through the duct.

Paralleling the study of wind velocity gradients, the Laboratory made a study of film conductance coefficients for a rough sanded surface and a smooth pine surface in a 12-in. and in a 6-in. duct. In these studies the object was not so much to determine usable coefficients for these conditions as it was to study the effect of the character of the wind velocity and position of its measurement on the coefficient which might be obtained. As a result of these studies a paper entitled Wind Velocity Gradients Near a Surface and Their Effect on Film Conductance is scheduled for presentation at the 1931 annual meeting.

d. Effect of Aging on the Conductivity of Concrete

This study has been under way at the Laboratory for the past three years during which time occasional tests have been made of the conductivities of two slabs of concrete of different mixes. These studies have shown a very high conductivity for concrete which is in keeping with a number of observations made in recent years both at the Laboratory and elsewhere. The study is further demonstrating the fact that the conductivity of concrete decreases with aging.

e. Absorption of Solar Radiation by a Roof or Wall Surface and the Effect of Heat Capacity of the Structure on Heat Penetration

For the past several years the Laboratory has been accumulating data on the effect of solar radiation on heat penetration into a building during the summer. This information is of vital importance to the air conditioning engineer. The early work on this subject gave information on the heat penetration through the lower surface of a roof or inner surface of a wall during warm summer days, and the variation in this rate of penetration throughout the 24-hour cycle. Early in 1930, the Laboratory presented data on the rate of heat absorption by the outside surface of a roof or wall depending upon the temperature and character of the surface, its angle, with the

direction of the sun's rays and other factors including data on the absorption of heat from solar rays by a plate of glass through which the sun shines.

More recently the Laboratory undertook a study of both the absorption of heat at the outside surface and its penetration through three roof panels for certain hot, sunny days in July, August, and September. This information is not yet sufficiently complete to warrant a conclusive technical paper on the subject. However, it is of interest to note the following facts demonstrated by the study: *First,* a horizontal black roof surface may reach a temperature as high as 170 F around noon on a clear July or August day due to the impingement of solar radiation on it. *Second,* this high temperature is maintained for only a comparatively short period of time at noon. *Third,* the rate of heat transfer through the structure never reaches the maximum which one would calculate from the maximum temperature difference between the top and bottom surfaces.

The greater the heat capacity of the structure the greater the difference between the maximum calculated rate of heat transfer and the measured rate. There is always a considerable lag between the maximum rate of heat penetration through to the top surface and through the bottom surface of the structure which lag increases with the increase in heat capacity of the structure. For a structure equivalent to 30 in. of concrete this lag amounts to approximately 15 hours so that the maximum rate of heat penetration into a building of this construction due to the maximum intensity of the sun around noon does not occur until early the next morning, while the minimum rate of penetration into such a building due to the cool of the preceding night takes place about the middle of the afternoon. The work so far carried on permits of only a qualitative statement of these magnitudes. However, additional work which will be continued through the coming year will probably make it possible to calculate heat flow through any roof or wall structure due to the impingement of the sun on the outside for any time of day as accurately as the conductivity, specific heat and density of the material in the structure and the intensity of the sun can be evaluated.

5. Study of Methods of Evaluating the Efficiency of Air Cleaning Devices.—*Technical Advisory Committee:* O. W. Armspach, *Chairman;* Albert Buenger, H. C. Murphy, Philip Drinker, C. A. Booth.

Early this year Professor Rowley presented data on the effect of certain characteristics of design of an impinger type of dust counter on the consistency of results which might be obtained with different instruments and different observers. This study showed conclusively that the variation in operating the pump on this type of instrument by different observers resulted in a variation in impingement of the air jet upon the plate receiving the dust sample which in turn resulted in a variation in characteristics of the sample collected for count. It was also shown that other factors such as the location of the orifice with respect to its distance from the slide and the size of the orifice resulted in inconsistent determinations. With the completion of this phase of the study the project has remained dormant pending the development of instruments in which these variable factors will be eliminated.

The present activity in connection with this study is aimed towards the development of standard specifications for an impinger type of instrument for use in dust counts for testing air cleaning devices. With the proper instrument available the next step will be to write a code for testing air cleaning devices.

6. Radiation.—*Technical Advisory Committee:* A. P. Kratz, *Chairman;* S. R. Lewis, R. S. Franklin, H. F. Hutzel, J. F. McIntire.

At the 1930 annual meeting of the Society it was agreed that the Research Committee might engage in the testing of radiators or similar devices for associations. The *Heating and Piping Contractors National Association* immediately presented a request that the Laboratory give an estimate of the possible cost of testing a large number of cast-iron radiators.

This report was given consideration by the Research Committee and the Technical Advisory Committee on Radiation, and it was agreed that the Laboratory would make such tests for the *Heating and Piping Contractors National Association* and a plan for

carrying out the work was developed including specifications for a test room which was to be built in Pittsburgh. The *Heating and Piping Contractors National Association* was advised that the Laboratory would undertake the work and would be ready to make the test within a month after it was certain that radiators would be provided. This study awaits further development.

7. Infiltration.—*Technical Advisory Committee:* G. L. Larson, *Chairman;* J. G. Shodron, M. S. Wunderlich, C. C. Schrader, J. E. Emswiler, D. S. Boyden.

During 1930 the Laboratory continued the study of air leakage through certain walls in the available apparatus in Pittsburgh and as a result published a paper entitled Air Leakage Through Various Forms of Building Construction which appeared in the December 1930 issue of the JOURNAL of the Society.

This paper, which was presented at the 1931 Annual Meeting, contains data on the following types of construction which will be of value in further improving the infiltration data in THE GUIDE.

1. A 13-in. brick wall without plaster.
2. The same wall after the application of plaster.
3. The same wall after the plaster had been removed and the joints repointed.
4. The same wall after the application of 1, 3, and 6 coats of paint to the brick.
5. A frame wall consisting of siding, paper, sheathing, stud, lath, and lime plaster.
6. A frame wall consisting of siding, paper, sheathing, stud, lath, and gypsum plaster.
7. The same as No. 6 with the application of sizing and paper to the plaster.
8. A frame stucco wall consisting of ⅞-in. sheathing, studding, paper, lath, and stucco.
9. A 13-in. brick and hollow tile wall.

The paper also contains data on air leakage through a crack between plaster and a door or window frame or baseboard.

During the first part of the year Professor Larson continued the study of air leakage through brick and frame walls at the University of Wisconsin in co-operation with the Laboratory. As a result of this study a paper entitled Air Infiltration through Various Types of Wood Frame Construction was published in the June 1930 issue of the JOURNAL of The Society and presented at the 1930 semi-annual meeting. This paper contained data on a large number of types of frame construction and will be of immense value in perfecting the infiltration section of THE GUIDE.

Professor Larson is now studying certain phases of air leakage around weather-stripped windows and the general subject of air drift across a building due to leakage in through the windows on the windward side, and leakage out through the windows, on the leeward side of the building.

8. Atmospheric Dust and Smoke.—*Technical Advisory Committee:* A. S. Langsdorf, *Chairman;* Dr. E. V. Hill, H. C. Murphy, Dr. S. W. Wynne, O. W. Armspach.

The study of atmospheric dust and smoke is a new venture with the Laboratory and was undertaken through a co-operative agreement between Washington University, St. Louis, and the Research Laboratory. According to this agreement, Dean Langsdorf is making a study of dust counting devices including a special scheme which he has for determining the dustiness of air by a light and electrical method. Dean Langsdorf is now undertaking this work but no results have yet been reported.

9. Air Conditions and Their Relation to Health.—*Technical Advisory Committee:* W. H. Carrier, *Chairman;* A. C. Willard, Dr. E. V. Hill, W. A. Rowe, C. P. Yaglou.

In this study which has been under way at the Laboratory for a number of years, work was continued on the determination of heat and moisture from the bodies of men working in still and moving air, and the heat loss from the bodies of children of school age. These phases of the study are now practically completed. A report of the heat and moisture loss from the bodies of men working was prepared and submitted to the *American Journal of Hygiene* and has since been approved for publication early in 1931.

The publication of Laboratory reports bearing on health and physiological subjects

in physiological and public health journals was approved recently by the Research Committee, it being thought advisable to get this material into the hands of the medical profession before publishing it in an engineering journal. After the article appears in the medical journal an engineering paper will be presented to the Society.

Additional plans for the future in connection with this study include a recheck of the comfort lines and a study of the effect of radiation to cold walls on the feeling of warmth.

Mr. Yaglou and Professor Drinker of the Harvard School of Public Health in co-operation with the Research Laboratory have been working during the past two years on the relation between the ionized condition of the atmosphere and ventilating conditions in a building. A paper on this subject is promised for the near future.

10. Measurement of Air Flow Through Registers and Grilles. —*Technical Advisory Committee:* S. R. Lewis, *Chairman;* John Aeberly, J. J. Haines, L. E. Davies, A. C. Willard.

The study of methods of measuring air flow through registers and grilles is being carried on by Professor Davies of Armour Institute of Technology, Chicago, in co-operation with the Research Laboratory. The purpose of this study is to standardize methods of measuring air flow through registers and grilles. The problem was brought to the Laboratory by *The Ventilating Contractors Employers Association* of Chicago and the Illinois Chapter of the Society. Professor Davies undertook the work a little over a year ago and completed one phase of the problem resulting in the paper entitled Measurement of Flow of Air Through Registers and Grilles which was published in the January 1930 issue of the JOURNAL.

Since making this report Professor Davies has continued the study of other phases of the subject and will have another report ready early in 1931. Recently John Howatt, Chairman of a Council Committee which is developing a Code for Testing Unit Ventilators, called upon the Laboratory for data applicable to the measurement of air delivery from such units.

11. Study of Methods of Testing Oil Burner Devices.—*Technical Advisory Committee:* L. E. Seeley, *Chairman;* H. F. Tapp, P. E. Fansler, H. R. Linn, J. H. McIlvaine, R. V. Frost.

This study is being carried on by Professor Seeley of Yale University in co-operation with the Research Laboratory. The *American Oil Burner Association* is co-operating with the Research Laboratory in connection with this study and is contributing largely towards the cost of the investigation. The investigation has for its purpose the development of a standard code for testing oil burner, boiler, or furnace combinations. For the purpose of the study, several boilers and burners so arranged that the burners can be transferred from one boiler to another have been installed and tested under different conditions. As a result of this study a Code for Testing Steam Heating Boilers Burning Oil Fuel was developed which is presented to the Society for approval.

President Harding referred to a matter of importance, namely, a request from various quarters to the officers of the Society for a revision of Section I of the Code of Minimum Requirements for the Heating and Ventilation of Buildings, which section deals with the subject of ventilation. He pointed out that this section is covered satisfactorily so far as engineers are concerned, but that apparently it is not presented in a satisfactory manner for legislative bodies.

Mr. Harding stated that the opinion had been expressed that it should be the function of a special committee to revise this section in co-operation with the various organizations that might be interested. He revealed that a number of state legislatures are at the present time proposing a revision of their ventilating

codes for public buildings, schools, etc., and asked for an expression of opinion on the subject, and possibly a motion.

W. H. Driscoll informed the audience that he had been appointed as a member of the committee to revise the Code of Minimum Requirements, and said that there is a mistaken opinion as to adaptibility of the Code of Minimum Requirements for legislative purposes, if used in toto. He did not feel that it was the function of the Code of Minimum Requirements Committee, nor did it have the proper personnel to handle the problem brought up by Mr. Harding. As to the legislative requirements of the code, Mr. Driscoll expressed the opinion that the committee must understand and have contact with problems of legislation both in the state and in the municipality, and suggested that the incoming President appoint a special committee with the idea of utilizing the information contained in the present Code of Minimum Requirements, and preparing it for purposes of legislation both for state and municipal codes.

This suggestion Mr. Driscoll submitted in the following motion, which was seconded:

Resolved That, the incoming president appoint a special Committee on Ventilation Standards which would utilize the information contained in the present Code of Minimum Requirements as a basis for the preparation of a Code which could be used for legislative purposes both by states and municipalities.

W. H. Carrier expressed the thought that the Code of Minimum Requirements as now written is an engineer's code, and is admirable as such, but thought that there should be more specific recommendations which would be usable to the layman and more understandable to him.

The motion submitted by Mr. Driscoll was adopted by unanimous vote.

F. C. McIntosh, Chairman of the Committee on Rules of Award for the F. Paul Anderson Award, announced the rules under which the first award would be made in January, 1932. He explained that this award was founded by Thornton Lewis at the 36th Annual Meeting of the Society in Philadelphia, January, 1930. At his request, the Council appointed a committee to prepare regulations governing this award. One of the stipulations is that the conditions governing any particular award shall be announced one year in advance. The Council has voted to give the F. Paul Anderson Award in 1932 to a member of the Society for outstanding service to the art of heating and ventilating.

The next item of business was the presentation of resolutions which were prepared by a committee of which J. F. Hale was chairman. The following resolutions were adopted:

RESOLVED THAT it is the sense of this meeting that we express our appreciation to the Chamber of Commerce of the City of Pittsburgh for the cordial welcome extended our Society by its representative.

RESOLVED THAT the Secretary of our Association be requested to express to the hotel management our appreciation of the excellent accommodations and service rendered during our stay in Pittsburgh.

RESOLVED THAT the members here assembled express their appreciation to the Westinghouse Electric and Manufacturing Company for their very interesting and illuminating demonstration and the opportunity afforded the President and President Elect of our Society to broadcast our message over their KDKA network.

RESOLVED THAT we extend to the several manufacturers identified with our industry our sincere appreciation for the very delightful visitations, which they arranged for the entertainment of our members and guests.

RESOLVED THAT the members here assembled at the 37th Annual Meeting of the Society extend to the Pittsburgh Chapter and its Committee on Arrangements our congratulations on the very interesting and unique program provided for our entertainment.

RESOLVED THAT the Ladies' Committee of our 37th Annual Meeting be congratulated for the graceful manner in which they looked after the interests and comforts of their guests.

RESOLVED THAT the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS here assembled wish to congratulate the Program Committee, the Authors, and the Technical Secretary of the Society for the excellent papers prepared and presented at this 37th Annual Meeting.

E. C. Evans suggested a revision for Articles IV and V of the Constitution.

After some discussion of this suggestion, it was agreed that it be referred to a committee which had already been appointed to revise the Constitution and the By-Laws of the Society, W. T. Jones being Chairman of this committee.

The final matter of business was the installation of the newly elected officers. With Past Presidents Willard, Driscoll, Hale, Hart and Anderson as escorts, W. H. Carrier, *President,* F. B. Rowley, *1st Vice-President,* W. T. Jones, *Second Vice-President,* and F. D. Mensing, *Treasurer,* were presented and retiring President Harding turned over the gavel to President Carrier. The new members of the Council, E. K. Campbell, Kansas City, R. Farnham, Buffalo, E. H. Gurney, Toronto, and E. O. Eastwood, Seattle, were introduced and installed.

After the presentation of the officers, President Carrier expressed appreciation for the honor that had been conferred upon him, and stated that the Society is growing in importance from year to year. He expressed the hope that it would continue to progress as it did under the administration of Mr. Harding.

In closing, Mr. Harding thanked the members of the Society and the officers who were associated with him during his administration for their wholehearted support.

PROGRAM 37TH ANNUAL MEETING

AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS

WILLIAM PENN HOTEL, PITTSBURGH, PA.

JANUARY 26-29, 1931

TECHNICAL PROGRAM

First Session, Monday, January 26

2:00 P.M. Greeting
Response—President L. A. Harding
Report of Council
Report of Secretary
Progress of Heating and Ventilating during the Past Quarter Century, by R. P. Bolton
Developments in Heating and Ventilating during 1930, by S. R. Lewis
Heat Emission from the Surfaces of Cast Iron and Copper Cylinders Heated with Low Pressure Steam, by A. C. Willard and A. P. Kratz
Heating Effect of Radiators, by R. V. Frost

Report of Tellers
Committee on Code for Testing and Rating Concealed Gravity Type Radiation by R. N. Trane, *Chairman*

Second Session, Tuesday, January 27

9:30 A.M. Report of Publication Committee—Prof. G. L. Larson
Development of the Ventilation System of the Holland Tunnel, by A. C. Davis
Noise Reduction in Ventilating Units, by Warren Ewald.
Noise and Ventilation, by G. T. Stanton
Report of Guide Publication Committee—D. S. Boyden
Report of Membership Committee—John D. Cassell
Report on Increase of Membership—C. W. Farrar

Third Session, Tuesday, January 27

2:00 P.M. Some Studies on the Absorption of Noise in Ventilating Ducts, by G. L. Larson and R. F. Norris
Report of Finance Committee—F. C. McIntosh
Air Conditioning as Applied to Furniture, Fixtures and Other Interior Woodwork, by D. R. Brewster
Air Conditioning in the Bakery, by W. L. Fleisher
Air Conditioning for Railway Cars, by A. H. Candee
Report of Committee on Oil Burning Devices, by L. E. Seeley, *Chairman*

Fourth Session, Wednesday, January 28

9:30 A.M. Welcome—O. P. Hood, Acting Director Bureau of Mines
Greeting—Dr. T. F. Baker, President Carnegie Institute of Technology
Report of Research Committee—Prof. F. B. Rowley, *Chairman*
Report of the Research Director—F. C. Houghten
Air Leakage through Various Forms of Building Construction, by F. C. Houghten, Carl Gutberlet and C. A. Herbert
Capacity of Return Mains for Steam and Vapor Heating Systems, by F. C. Houghten and Carl Gutberlet
Welded Piping for Building Heating Systems, by F. G. Outcalt
Air Pollution from the Engineers' Standpoint, by H. B. Meller

Wednesday, January 28

2:00 P.M. Inspection Trip—Research Laboratory, Bureau of Mines and Carnegie Institute of Technology

Fifth Session, Thursday, January 29

9:30 A.M. Smoke and Dust Abatement, by M. D. Engle
Spectral Distribution of the Energy Radiated from Metallic Surfaces at High Temperatures, by G. R. Greenslade
Insulating Effect of Successive Air Spaces Bounded by Bright Metallic Surfaces, by L. W. Schad
Wind Velocity Gradients Near a Surface and Their Relation to Film Conductance by F. C. Houghten and Paul McDermott

Sixth Session, Thursday, January 29

2:00 P.M. Utilization of Hot Springs for Heating in Iceland, by Ben Gröndal
Off-Peak System of Electric Heating, by Elliott Harrington
Heating with Exhaust Steam, by Perry West
Installation of Officers
Announcement of Rules of Award for F. Paul Anderson Medal
Resolutions
Adjournment
4:30 P.M. Council Meeting (Wm. Penn Hotel)

ENTERTAINMENT

Monday, January 26

12:30 P.M. Luncheon for Council and Authors
2:00 P.M. Inspection Tour and Tea for Ladies at Kaufman's Department Store
9:00 P.M. A Night in Pirate Hall (William Penn Hotel) Westinghouse Program, Music, Cards, Dancing.

Tuesday, January 27

8:00 A.M. Breakfast Meeting—Committee on Research
1:00 P.M. Luncheon and Bridge for Ladies—Crystal Room, William Penn Hotel
6:30 P.M. Annual Dinner Past Presidents
Dinner for Wives of Past Presidents
8:00 P.M. Miniature Championship Golf Tournament for Miniature Research Cup (William Penn Miniature Course)

Wednesday, January 28

10:45 A.M. Sightseeing trip for Ladies
Lunch at Heinz
1:00 P.M. Meeting of Nominating Committee—Pittsburgh Athletic Club
1:15 P.M. Luncheon for Members—Pittsburgh Athletic Club
7:30 P.M. Annual Banquet and Dance—Urban Room, William Penn Hotel

Thursday, January 29

Visit to KDKA Broadcasting Station William Penn Hotel
Inspection of Industrial Plants

COMMITTEE ON ARRANGEMENTS

H. LEE MOORE, *Chairman*

H. B. ORR, *Chairman*, Reception Committee.
F. C. HOUGHTEN, *Chairman*, Program Committee.
R. B. STANGER, *Chairman*, Entertainment Committee.
F. A. GUNTHER, *Chairman*, Banquet Committee.
W. W. STEVENSON, *Chairman*, Transportation Committee.
J. E. MCGINNESS, *Chairman*, Publicity Committee.
F. C. MCINTOSH, *Chairman*, Finance Committee.

No. 878

A.S.H.V.E. STANDARD CODE FOR TESTING STEAM HEATING BOILERS BURNING OIL FUEL

COMMITTEE: L. E. Seeley, *Chairman*, P. E. Fansler, R. V. Frost, H. R. Linn, J. H. McIlvaine, H. F. Tapp and F. C. Houghten

I. INTRODUCTION

1. **The purpose** of the Code for Testing Steam Heating Boilers Burning Oil Fuel is to provide a standard method for conducting and reporting tests to determine heat efficiency and performance characteristics. The Code recognizes that tests of oil burners and boilers may be made under different conditions for different purposes, and with complete or limited facilities for conducting the tests. It is designed to cover the determination of a complete heat balance; if a less complete test is required to satisfy the objectives of the test, observations not required may be omitted. It should be appreciated that in all cases to which this code applies there are two devices (that is, boiler and oil burner) involved in the test. Both devices influence the results obtained. Two different oil burners placed successively in the same boiler may not produce identical results. To view the results as if they pertained only to the boiler, would be an incorrect viewpoint.

2. **Standard Form.** Table 1, Standard Form, Oil Fuels, is arranged for recording the full data called for by this Code.

3. **Short-Test Form.** Table 2, Short-Test Form, Oil Fuels, is for tests in which a complete heat balance is not desired. The following determinations are not required:

(a) Ultimate analysis of fuel
(b) Complete flue gas analysis
(c) Furnace volume *A* and *B*
(d) Unit combustion rate

4. **Continuous and Intermittent Tests.** Two general types of tests are recognized which may use either the Standard or Short-Test Form. These will be designated as Continuous and Intermittent. A continuous test may be defined as one where fuel is burned throughout the duration of the test at a constant rate. An Intermittent test is one where the fuel may not be burning at all times during the test or, if burning continuously, the rate of burning varies. These methods are treated in detail in Sections VII and VIII of the Code.

II. DEFINITIONS

5. **Furnace Volume.** There are two furnace volumes to be considered, which will be designated as furnace *A* and furnace *B*.

6. **Furnace A** is the total volume enclosed by the boiler and boiler foundation which may be used as combustion space. This space is that enclosed by the floor,

Report of Cooperative Research of the A.S.H.V.E. Research Laboratory and the *American Oil Burner Association* conducted at Yale University prepared by A.S.H.V.E. Technical Advisory Committee on Oil Burning Devices and presented at the 37th Annual Meeting of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, Pittsburgh, Pa., January 1931.

side walls, and/or water legs, up to the crown sheet or equivalent which shall be a plane tangent to the bottom of the lowest row of tubes or other water backed surfaces. A bridgewall shall constitute a side wall. The average width, length and height of the furnace shall be indicated on a dimensioned sketch.

7. **Furnace B** is the space left in furnace *A* after the oil burner has been installed. The lower bounding surface will generally be the hearth of the furnace or equivalent, the side bounding surface the same as furnace *A*, except as modified by brick work on the sides and ends, and the upper bounding surface the same as furnace *A*. Again the average dimensions shall be recorded on a sketch.

8. **Combustion Rate.** This shall be recorded as the number of gallons, pounds and Btu per hour supplied to the burner.

9. **Unit Combustion Rate.** This shall be recorded as the number of gallons, pounds and Btu per hour per cubic foot of furnace *B* supplied to the burner.

10. **Boiler Output.** The quantity of heat available at the boiler nozzle with the boiler normally insulated.

11. **Overall Efficiency.** The percentage of the heat in the oil fuel fired which is delivered at the boiler nozzle when the boiler is normally insulated.

III. SET-UP OF OIL BURNER AND BOILER TO BE TESTED

12. **The Oil Burner.** If the oil burner is set up for the purpose of commercial testing, it shall be installed according to manufacturer's directions.

Note: Experimental work is limited in no manner except that the manner of installation shall be clearly described.

13. **The Boiler.** If the boiler is set up for the purpose of testing, it shall be assembled according to manufacturer's directions.

14. **Boilers** already installed shall be arranged to comply with the following conditions as nearly as possible, or as shall be agreed to by all interested parties.

15. **Boiler Covering.** The boiler should be covered with such heat insulation as is recommended by the manufacturer; if tested uninsulated or partly insulated an allowance shall be made for that portion of the heat lost from the bare iron steam and water surfaces which might have been saved by covering; the allowance shall be computed from the formula given with the tables of this Code.

16. **Steam Outlet Piping.** All steam outlet pipes from the boiler shall always be well covered, up to and including the steam separator.

17. **The vertical outlets** from the boiler leading up to the separator shall be in number, arrangement and size according to manufacturer's directions and shall rise vertically from the boiler nozzle as high as possible but not more than 30 in. before entering the steam separator.

18. **The steam piping** leading from the separator shall have a slope so that water condensed in it will drain away from the separator. The drain pipe from the steam separator shall include a *U*-leg sufficiently long to prevent the steam blowing out, and with a short inverted *U*-leg at the outlet, so that the drip may be caught in a container. Re-evaporation of water drained from the separator shall be prevented by use of a suitable cover on the container.

19. **The steam valve** on the boiler steam outlet piping shall be placed beyond the steam separator

20. **Feed Water Piping.** The feed water pipe should connect to the boiler where the returns would normally come. The temperature of the feed water shall be read from a thermometer inserted in a cup projecting well into the feed line near the boiler and filled with a heavy oil or mercury. A valve shall be placed on each side of the above-mentioned cup. The valve between the cup and boiler shall serve as a shut-off valve. The valve between the cup and feed water supply shall be used as a regulating valve only. This valve may be an automatic one for maintaining a uniform water level in the boiler.

21. **All boiler-water connections,** including blow-off pipes, must be exposed to view, so that leakage may be observed and either stopped or measured. It is de-

sirable to have a steady rate of feed so that feed water temperature readings will be accurate and none will have to be omitted.

22. **Water Gage Marking.** The water gage shall have a mark to indicate the water level recommended by the manufacturer.

23. **Smokehood Connections.** The boiler shall be connected with a short, direct smoke pipe to a chimney flue of suitable size, height and construction to give proper draft. All joints shall be thoroughly sealed and maintained so during the tests.

24. **Insulation.** The smokehood and the smoke flue shall be insulated with not less than 1-in. thickness of insulation; the smoke-flue insulation shall extend at least one equivalent flue diameter beyond the point where the flue-gas temperature is measured.

25. **The choke damper,** if incorporated in the boiler, shall be put in place, opened wide and kept so during all tests. A damper shall be placed in the smoke pipe between the chimney and the point where flue-gas measurements are made. This shall be called the regulating damper. A check damper, either manual or automatic, may be placed between this damper and the chimney if desired.

26. **Chimney.** This code allows that the draft may be produced by a chimney or by a fan or other arrangement for providing induced draft; there shall, however, be a suitable damper or by-pass in the test flue or chimney that will permit of the draft at the regulating stack damper being reduced to a little above that required for the output of the test which is being made.

27. **Cleaning of Boiler.** The water spaces of the boiler shall be thoroughly boiled out with a cleansing solution, then thoroughly rinsed with clean water.

28. **Soot and Dust.** The heating surface, furnace, flues and chimney shall be clean and free from soot and dust at the beginning of the test.

IV. INSTRUMENTS AND MEASURING APPARATUS

29. **Steam Output.** The weight of water evaporated may be determined either by condensing the steam and weighing the condensate, designated as Method *A*, or by weighing the water fed to the boiler, designated as Method *B*.

30. **Method A.** Water shall be fed to the boiler without being weighed. The steam after passing through the separator shall be led to a condenser through a connecting steam pipe sloping toward the condenser. The condensate shall be caught in suitable tanks and weighed. The condenser shall be the closed type, preferably with outside joints, and of sufficient capacity that the temperature of the condensate shall not exceed 150 F. Before each test it shall be tested for freedom from leakage by putting it under full water pressure.

31. **Method B.** The quantity of water fed to the boiler shall be determined in weighing or measuring in a calibrated tank. The most convenient weighing arrangement depends on the size of the boiler, but it shall be such that the weight fed can be determined at any instant. The use of tanks calibrated for the weight of water they hold at the temperature of the feed is permissible. The water may be fed to the boiler by gravity, air pressure or feed pumps; there shall be no leakage between the measuring tanks and the boiler.

32. **Weighing Scales.** Accurate scales of suitable size shall be provided for weighing separator water, feed water, and fuel. If the oil tank is mounted on scale it shall be connected to the burner by means of flexible oil-tight tubing. Water and oil may be volumetrically measured by means of calibrated tanks.

33. **Draft.** Two draft gages shall be provided and so arranged as to determine the pressure difference between the room atmosphere and the furnace and the smokehood. Draft measurements shall be made with draft gages reading to 0.01 in. Draft gages shall be checked for zero reading each hour.

34. **The smokehood draft** shall be measured at least one pipe diameter before the regulating stack damper. The connection into the stack flue shall have its end square and shall be set at right angles to the flow of the gases. A ⅛-in. pipe is a convenient size.

35. Steam Separator. The steam separator shall have sufficient volume to catch slugs of water and to prevent their passing into the steam lines; it shall also efficiently separate entrained water from steam carrying it.

36. Temperature Measurement. Accurately calibrated instruments shall be provided for all temperature measurements. A mercury thermometer is preferable for measuring feed water temperature. It is recommended that thermocouples, in preference to other types of thermometers, be used to measure the flue-gas temperatures. The temperature shall be measured in the smoke pipe not less than one nor more than two pipe diameters beyond the smoke-pipe collar. The thermo-junction or bulb of the thermometer shall be placed as shown on the sketch. The instrument used with the couple should be sensitive to at least 5 deg Fahrenheit and have a range up to 1200 F.

37. Gas Analysis. The sample for gas analysis shall be taken from the smoke pipe at the point at which the temperature is measured in such a manner as to give a fair average sample of the gas stream. An Orsat or equivalent gas analyzer shall be used. An open end ⅛-in. pipe reaching one-quarter way across the flue pipe is usually satisfactory, but the probable mixing of the gases should be considered and the open end so placed as to be in the average gas stream. For gas temperatures of over 750 F the portion of the collecting pipe extending into the flue should be clay or silica. As an average value for the test period is desired, the correct and simplest method is to collect samples of the flue gas at a constant rate into a

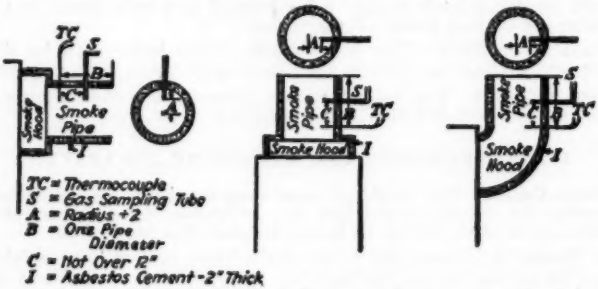

FIG. 1—LOCATION OF GAS TEMPERATURE THERMOCOUPLE AND GAS SAMPLING TUBE

one-gallon bottle approximately, and to take the samples for analysis from these bottles. Each of such samples shall be collected for the same length of time and their analyses averaged. If recording gas analyzing instruments are provided, they shall be checked every hour with the Orsat or other manual apparatus.

38. Smoke Readings. Smoke readings if desired shall be made by the Ringelmann Chart Method.

39. Pressure Readings. A calibrated steam gage or a mercury column shall be used for determining the steam pressure.

V. FUEL SAMPLING

40. A fair sample of the oil shall be obtained (not less than two (2) quarts). This may be done by taking some oil from the small tank especially used for testing or an occasional sample from the oil supply line to the burner. The sample should be placed in a tight vessel, marked with test number and stored in a cool place.

41. Fuel analysis. The fuel shall be analyzed according to the method of test specified by the *American Society for Testing Materials.*

Note: For the Standard form the ultimate analysis must be obtained in addition to the usual findings. The short form requires only a heat value determination.

42. A calculated heat value may be used, especially in field work, with reasonable accuracy for that type of work. It is not recommended for laboratory tests (see item 17, Table 2).

VI. DURATION OF TESTS

43. The duration of tests shall be based on the firing of the minimum quantity of fuel oil to make the quantitative errors comparable for all tests.

44. For the continuous test the total quantity of oil fuel burned shall be at least two pounds of oil per cubic feet of Furnace *B* (see par. 5) and under no condition shall the test be less than two hours duration.

45. For the intermittent test the total quantity of oil fuel burned shall be at least three pounds of oil per cubic foot of Furnace *B* (see par. 5).

VII. STARTING AND STOPPING CONTINUOUS TESTS

46. Operating Conditions. After the burner has been started and the combustion rate and draft have been adjusted, all operating conditions shall have remained constant for one-half hour before the test is started.

47. Start of test. At the start of the test the time, fuel, feed water and gage glass level shall be quickly measured and recorded. A pail shall be placed under the separator outlet.

48. End of test. At the end of the test the gage glass level should be at exactly the same point existing at the beginning of the test and the time, fuel and feed water shall be quickly measured and recorded. The separator water in the pail shall be weighed and recorded.

VIII. STARTING AND STOPPING OF INTERMITTENT TESTS

49. Measurements and procedure. In general the same measurements and procedure must be followed in these tests as outlined in Section VII. Due to the fact that intermittent operation includes a heating and cooling of the boiler, firebrick and so forth, it is absolutely necessary to be sure that the same conditions exist at both the beginning and end of the test.

50. The duration of the *on* or burning period of the test shall be such that conditions have become reasonably stabilized before going to the *off* or *non-burning* or possible *low-burning* period of the test. Flue gas temperature will show this stabilized condition quite well if succeeding readings show or indicate but small changes. For small boilers 30 min. seems long enough. For large boilers longer periods might be necessary.

51. These tests shall be started and stopped at the end of an *on* period of sufficient duration as explained in par. 50. For any one test all the *off* periods shall be equal to each other in duration. The test readings except time and draft may be omitted during the *off* period except in cases where fuel is burning. To get correct averages when operating on the *low-burning* period it is necessary to obtain weighted averages of the flue-gas temperature and analysis readings according to the respective rates of fuel burning. This will give the required accuracy on the heat balance calculations called for in the Standard Test Form.

IX. METHOD OF CONDUCTING TEST

52. The steam pressure shall be maintained at 2 lb unless the purpose of the test requires operation at some other definite pressure. The water level in the boiler shall be maintained constant during the test.

53. Water fed or condensed. The records of water fed or condensed shall be made as required by the measuring system. It is desirable to measure the water fed or condensed each half hour to check the rate of steaming.

54. Observations of drafts, pressures and temperatures shall be made regularly during the test at least every 15 min. Separator water shall be weighed every hour.

55. Flue gas analyses shall preferably be made on samples collected for periods not exceeding 30 min.

56. Smoke readings when made shall be recorded at least every 30 seconds over a period of time sufficient to show the performance under all typical conditions. See Section IV, Par. 38.

X. PERFORMANCE CHARTS

57. If four or more continuous tests are run at different combustion rates a performance chart may be made. Against output expressed in thousands of Btu and square feet of steam radiation, the following items may be plotted; overall efficiency, flue temperature, combustion rate, furnace draft, boiler draft, unit combustion rate and CO_2.

58. If four or more intermittent tests are run at different *off* periods a performance chart may be drawn plotting overall efficiency against output expressed in thousand Btu and square feet of steam radiation.

STANDARD FORM

TABLE 1. DATA AND RESULTS OF TEST—LIQUID FUELS

General Information

1. Number of test .....
2. Date of test .....
3. Test conducted by .....
4. Type of test: continuous.....intermittent.....
5. Make and catalogue designation of burner.....
6. Type of burner .....
7. Kind of ignition .....
8. Make and catalogue designation of boiler .....
9. Type of boiler .....
10. Description of boiler covering .....
11. Heating surface of boiler .....sq ft
12. Furnace volume *A* .....cu ft
13. Furnace volume *B* .....cu ft
14. Weight of water in boiler to normal water line.....lb
15. Duration of test .....hours
16. Power consumed by oil burner .....watts
17. Gas consumed by gas pilot.....cu ft
18. Oil setting (valve position, oil pressure, nozzle, etc.).....
19. Air setting (number of fan and adjustment, area of air opening, air pressure, etc.) .....
20. Description of flame (color, shape, condition, etc.) .....
21. Barometer reading .....in. mercury

Fuel

22. Kind (Commercial Standard Grade) Number .....
23. Gravity .....°A.P.I.
24. *Flash point .....F
25. *Distillation Test (*a*) 10% at .....F
    (*b*) 90% at .....F
    (*c*) E.P. at .....F
26. *Water and sediment .....per cent
27. **Viscosity ..... seconds ..... method at ..... F
28. Ultimate analysis:
    (*a*) Hydrogen .....per cent of total weight
    (*b*) Carbon .....per cent of total weight
    (*c*) Nitrogen .....per cent of total weight
    (*d*) Oxygen .....per cent of total weight
    (*e*) Sulphur .....per cent of total weight
29. (*a*) Higher Heating value per pound as fired .....Btu
    (*b*) Higher Heating value per pound moisture free.....Btu
30. Total fuel fired during test .....lb
31. Temperature of oil as fired .....F

* According to Commercial Standard Specifications.
** Saybolt Universal at 100 F for No. 3 and No. 4 oil.
Saybolt Furol at 122 F for No. 5 and No. 6 oil.

Draft

32. Draft in furnace .....in. of water
33. Draft at boiler outlet .....in. of water

Flue Gas

34. Carbon Dioxide (CO_2) .....per cent volume
35. Oxygen (O_2) .....per cent volume
36. Carbon Monoxide (CO) .....per cent volume
37. Nitrogen (N_2) by difference .....per cent volume
38. Temperature of flue gases leaving the boiler.....F
39. Temperature of air for combustion.....F
40. Dry flue gas per pound of fuel, as fired.....lb
41. Air supplied per pound of fuel, as fired (actual).....lb
42. Theoretical air required per pound of fuel as fired.....lb
43. Excess air .....per cent

Steam and Feed Water

44. Steam pressure by gage, boiler.....lb per sq in.
44. Absolute steam pressure .....lb per sq in.
45. Temperature of feed water entering boiler.....F
46. Total weight of water fed to boiler.....lb
47. Total weight of water from separator.....lb
48. Percentage of moisture in steam.....per cent
49. Total water evaporated .....lb
50. Factor of evaporation .....

Hourly Rates

51. Combustion rate .....lb per hour
52. Combustion rate .....gal per hour
53. Combustion rate .....1000 Btu per hour
54. Unit combustion rate .....lb per cu ft of furnace B per hour
55. Unit combustion rate.....gal per cu ft of furnace B per hour
56. Unit combustion rate.....1000 Btu per cu ft of furnace B per hour
57. Actual water evaporated per hour.....lb per hour
58. Correction for uncovered boiler.....lb per hour
59. Total equivalent evaporation.....lb per hour
60. Heat absorbed per square foot of boiler heating surface per hour .....Btu per square foot per hour

Output

61. Output in 1000 Btu per hour..... .....Btu
62. Output in square feet of steam radiation.....sq ft
63. Actual evaporation per pound of fuel, as fired.....lb
64. Equivalent evaporation per pound of fuel, as fired.....lb

Heat Balance

65. Heat transferred to boiler (overall efficiency of boiler, burner and furnace) .....per cent Btu
66. Heat carried away by steam in flue gases.....per cent Btu
67. Heat carried away by dry flue gases.....per cent Btu
68. Heat loss due to unconsumed carbon monoxide.....per cent Btu
69. Heat loss due to unconsumed hydrogen, hydrocarbons, radiation and unaccounted for miscellaneous items.....per cent Btu

CALCULATIONS FOR TABLE 1

Explanation of Symbols

Wherever CO_2, CO, O_2, and N_2 are used they are percentages by volume of these constituents in the gases of combustion.

H = Total heat (in Btu per pound of dry saturated steam at the boiler outlet pressure.)

h = Total heat (Btu per pound) in feed water at boiler inlet.

L = Latent heat (in Btu per pound) of steam at the boiler outlet pressure.

Formulae and Notes

$$\text{Item 40} = \frac{700 + 4\,CO_2 + O_2}{3\,(CO_2 + CO)} \times \frac{\text{Item 28 } (b)}{100}$$

$$\text{Item 41} = \text{Item 40} + \frac{9 \times \text{Item 28 } (a)}{100} - 1$$

$$\text{Item 42} = 0.3456\left[\frac{(\text{Item 28 } (b)}{3} + \left(\text{Item 28 } (a) - \frac{\text{Item 28 } (d)}{8}\right) + \frac{\text{Item 28 } (e)}{8}\right] \text{ or}$$

$$11.52\,\frac{\text{Item 28 } (b)}{100} + 34.56\left(\frac{\text{Item 28 } (a)}{100} - \frac{\text{Item 28 } (d)}{100 \times 8}\right) + 4.32\,\frac{\text{Item 28 } (e)}{100}$$

$$\text{Item 43} = \frac{\text{Item 41} - \text{Item 42}}{\text{Item 42}} \times 100$$

Item 44*a* = (Barometric Pressure in Inches of Mercury × 0.491) + Item 44

Item 46 = (When condenser is used) weight of water from condenser + Item 47

$$\text{Item 48} = \frac{\text{Item 47}}{\text{Item 46}} \times 100$$

Item 49 = Item 46 − Item 47

$$\text{Item 50} = \frac{H - \dfrac{L \times \text{Item 48}}{100} - h}{971.7}$$

$$\text{Item 51} = \frac{\text{Item 30}}{\text{Item 15}} \qquad \text{Item 52} = \frac{\text{Item 30} \times (131.5 + \text{A.P.I.gravity})}{\text{Item 15} \times 1180}$$

$$\text{Item 53} = \frac{\text{Item 30} \times \text{Item 29 } (a)}{\text{Item 15} \times 1000} \qquad \text{Item 54} = \frac{\text{Item 51}}{\text{Item 13}}$$

$$\text{Item 55} = \frac{\text{Item 52}}{\text{Item 13}} \qquad \text{Item 56} = \frac{\text{Item 53}}{\text{Item 13}} \qquad \text{Item 57} = \frac{\text{Item 49}}{\text{Item 15}}$$

$$\text{Item 58} = \frac{1.8}{H - h}\,(T_s - \text{Item 39})\,A_s \textit{ where}$$

T_s = temperature of steam at the boiler pressure: A_s is the allowed projected area of uncovered metal steam-heated surface. This item shall not be allowed on a commercial acceptance test.

Item 59 = (Item 57 + Item 58) × Item 50

$$\text{Item 61} = \frac{(\text{Item 59} \times 971.7)}{1000} \qquad \text{Item 62} = \text{Item 61} \times \frac{971.17}{240}$$

$$\text{Item 63} = \frac{\text{Item 49}}{\text{Item 30}} \qquad \text{Item 64} = \frac{\text{Item 59} \times \text{Item 15}}{\text{Item 30}}$$

$$\text{Item 65} = \frac{\text{Item 64} \times 971.7 \times 100}{\text{Item 29 } (a)}$$

$$\text{Item 66} = \frac{9 \times [1089 + 0.46\,(\text{Item 38} - \text{Item 39})] \times \text{Item 28 } (a)}{\text{Item 29 } (a)}$$

$$\text{Item 67} = \frac{\text{Item 40} \times 0.24\,(\text{Item 38} - \text{Item 39}) \times 100}{\text{Item 29 } (a)}$$

$$\text{Item 68} = \frac{\dfrac{CO}{CO_2 + CO} \times \text{Item 28 } (b) \times 10{,}150}{\text{Item 29 } (a)}$$

Item 69 = 100 − (Item 65 + Item 66 + Item 67 + Item 68)

SHORT TEST FORM

TABLE 2. DATA AND RESULTS OF TEST—LIQUID FUELS

General Information

1. Number of test .....
2. Date of test .....
3. Test conducted by .....
4. Type of test: continuous.....intermittent.....
5. Make and catalogue designation of burner.....
6. Type of burner .....
7. Kind of ignition .....
8. Make and catalogue designation of boiler.....
9. Type of boiler .....
10. Description of boiler covering .....
11. Duration of test .....hours
12. Oil setting (valve position, oil pressure, nozzle, etc.).....
13. Air setting (Number of fan adjustment, area of air opening, air pressure, etc.)...
14. Description of flame (color, shape, condition, etc.).....

Fuel

15. Kind (Commercial Standard Grade) Number.....
16. Gravity .....°A.P.I.
17. Higher heating value per pound as fired (calculated) (calorimeter).....Btu
18. Total fuel fired during test.....lb

Draft

19. Draft in furnace .....in. water
20. Draft at boiler outlet.....in. water

Flue Gas

21. Carbon Dioxide (CO_2).....per cent volume
22. Oxygen (O_2).....per cent volume
23. Carbon Monoxide (CO).....per cent volume
24. Temperature of flue gases leaving boiler.....F

Steam and Feed Water

25. Steam pressure by gage boiler.....pounds per square inch
25a. Absolute steam pressure .....pounds per square inch
26. Temperature of feed water entering boiler.....F
27. Total weight of water fed to boiler.....lb
28. Total weight of water from separator.....lb
29. Per cent of moisture in steam.....per cent
30. Total water evaporated .....lb
31. Factor of evaporation .....lb

Hourly Rates

32. Combustion rate .....lb per hour
33. Combustion rate .....gal per hour
34. Actual water evaporated .....lb per hour
35. Correction for uncovered boiler.....lb per hour
36. Total equivalent evaporation.....lb per hour

Output

37. Output in 1000 Btu per hour.....Btu
38. Output in square feet of steam radiation.....sq ft
39. Actual evaporation per pound of fuel as fired.....lb
40. Equivalent evaporation per pound of fuel as fired.....lb
41. Heat transferred to boiler (overall efficiency of boiler and furnace).per cent Btu

CALCULATIONS FOR TABLE 2

Explanation of Symbols

H = Total heat (in Btu per pound) of dry saturated steam at the boiler outlet pressure.
h = Total heat (in Btu per pound) in feed water at boiler inlet.
L = Latent heat (in Btu per pound) of steam at boiler outlet pressure.

Formulae and Notes

Item 17 = Calculated heat value = 18650 + 40 (°A.P.I.) − 10)

Item 25a = (Barometric Pressure Inches of Mercury × 0.49) + Item 25

Item 27 = (When condenser is used) weight of water from condenser + Item 28

$$\text{Item 29} = \frac{\text{Item 28}}{\text{Item 27}} \times 100$$

$$\text{Item 30} = \text{Item 27} - \text{Item 28}$$

$$\text{Item 31} = \frac{H - \dfrac{L \times \text{Item 29}}{100} - h}{971.7}$$

$$\text{Item 32} = \frac{\text{Item 18}}{\text{Item 11}}$$

$$\text{Item 33} = \frac{\text{Item 18} \times (131.5 + \text{A.P.I. gravity})}{\text{Item 11} \times 1180}$$

$$\text{Item 34} = \frac{\text{Item 30}}{\text{Item 11}}$$

$$\text{Item 35} = \frac{1.8}{H - h}\,(T_s - T_a) \times A_s$$ where T_s = temperature of steam at the boiler pressure; T_a = temperature of air; A_s is the allowed projected area of uncovered metal steam-heated surface. This item shall not be allowed on a commercial acceptance test.

Item 36 = (Item 34 + Item 35) × Item 31

$$\text{Item 37} = \frac{\text{Item 36} \times 971.7}{1000}$$

$$\text{Item 38} = \text{Item 37} \times \frac{971.7}{240}$$

$$\text{Item 39} = \frac{\text{Item 30}}{\text{Item 18}}$$

$$\text{Item 40} = \frac{\text{Item 36} \times \text{Item 11}}{\text{Item 18}}$$

$$\text{Item 41} = \frac{\text{Item 40} \times 971.7 \times 100}{\text{Item 17}}$$

No. 879

HEAT EMISSION FROM THE SURFACES OF CAST IRON AND COPPER CYLINDERS HEATED WITH LOW PRESSURE STEAM

By A. C. WILLARD[1] AND A. P. KRATZ[2] URBANA, ILLINOIS

MEMBERS

This paper attempts to explain why a cast-iron cylinder with walls 0.25 *in. thick will transmit over* 50 *per cent more heat than a copper cylinder of exactly the same size with walls* 0.01264 *in. thick, when there is low pressure steam on the inside of the cylinders and air on the outside of the cylinders. Note that the cast-iron cylinder wall is almost* 20 *times as thick as the copper cylinder wall, and that copper has a conductivity approximately* 8 *times that of cast-iron per inch of thickness. Although the copper wall is* $8 \times 20 = 160$ *times better than the cast-iron wall as a heat conductor, the overall heat transfer of the cast-iron cylinder is greater than that of the copper cylinder because of the greater surface emission of the cast iron.*

THERE appears to be more or less confusion[3] in the minds of heating engineers and others concerning the transmission of heat through, and the emission of heat from metal surfaces *exposed to air on one side and steam on the other.* This paper presents the results of some recent simple tests on cast-iron and copper cylinders as an illustration of certain well-known laws of heat transmission and emission under temperature conditions comparable with those which exist in heating buildings.

The authors make no claim for originality or novelty in either the scientific facts or the following discussion based on these facts. Acknowledgment is due E. L. Broderick, Research Assistant in Mechanical Engineering, who conducted at the University of Illinois the recent tests referred to in the paper.

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[1] Professor of Heating and Ventilation and Head of Department of Mechanical Engineering, University of Illinois.

[2] Research Professor, Engineering Experiment Station, University of Illinois.

[3] Such statements as "radiators today are built *with thinner walls,* which allow a ready escape of the heat," taken from a paper which has just appeared in print, are common examples of this "confusion."

Presented at the 37th Annual Meeting of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, Pittsburgh, Pa., January, 1931, by A. C. Willard.

GENERAL STATEMENT

It is generally *assumed* that a thin layer of a given material will transmit more heat from the hot side, or surface, to the cool side, or surface, than a thicker layer of the same material, provided the hot surfaces are at the same temperature in both cases, and the cool surfaces are exposed to the same medium simultaneously.

It is also generally *assumed* that of two different materials of the same thickness, the material having the higher coefficient of conductivity will transmit more heat from the hot side, or surface, to the cool side, or surface, than the other material with a lower coefficient of conductivity, provided the hot surfaces are at the same temperature in both cases, and the cool surfaces are exposed to the same medium simultaneously.

There is a "joker" missing in both of the preceding assumptions, in that two important details or conditions have been omitted. These two details, which are commonly overlooked, are the type or character of the cool side or surface, and the nature or kind of medium on the cool side of the materials. Failure to keep these details in mind when considering the heat transmission of *metals* exposed to *steam* on the hot surface and *air* on the cool surface is the cause of the "confusion" referred to in the opening sentence of the paper.

So important are these details or conditions that, contrary to the generally *assumed* conceptions of heat transmission just presented, it may be readily demonstrated that a cast-iron cylinder 0.25 in. thick, painted a flat gray, will transmit 2.43 Btu per square foot per degree Fahrenheit per hour as compared with 1.41 Btu per square foot per degree Fahrenheit per hour for a clean copper cylinder of No. 28 gage metal (0.01264 in. thick), when both are supplied with steam at about 1 lb gage pressure, and the outer surfaces are exposed to air at about 75 F. Note that the walls of the cast-iron cylinder are almost 20 *times as thick* as the walls of the copper cylinder. Moreover, these apparently contradictory results exist in spite of the fact that the conductivity of copper is about 2,730 Btu per square foot per inch per hour per degree Fahrenheit, as compared with 325 Btu for cast iron. Note, therefore, that the conductivity of the copper cylinder walls is about 8 *times as great* as the conductivity of the cast-iron cylinder walls *per inch of thickness.*

TEST PLANT AND PROCEDURE

The simple tests already referred to were made with the same type of plant (Fig. 1) as that used in an earlier study [4] of the heat emission from various surfaces, using cylinders 10 in. in diameter and 20 in. in length, of copper, cast-iron and sheet iron. The plant as actually used in these recent tests is shown in Fig. 2 and has been in service for several years.

Saturated steam was supplied to the cylinders at a pressure of about 1½ in. mercury or 215 F, and the air surrounding the cylinders was approximately 75 F. Since all tests were run simultaneously in still air, slight variations in conditions were reflected equally in the performance of all cylinders.

A thermocouple of copper and constantan wire No. 34 B. and S. gage, was attached flush with the outer surface of the cast-iron cylinder and located

[4] Emissivity of Heat from Various Surfaces, by V. S. Day (Engineering Experiment Station Bulletin No. 117, University of Illinois).

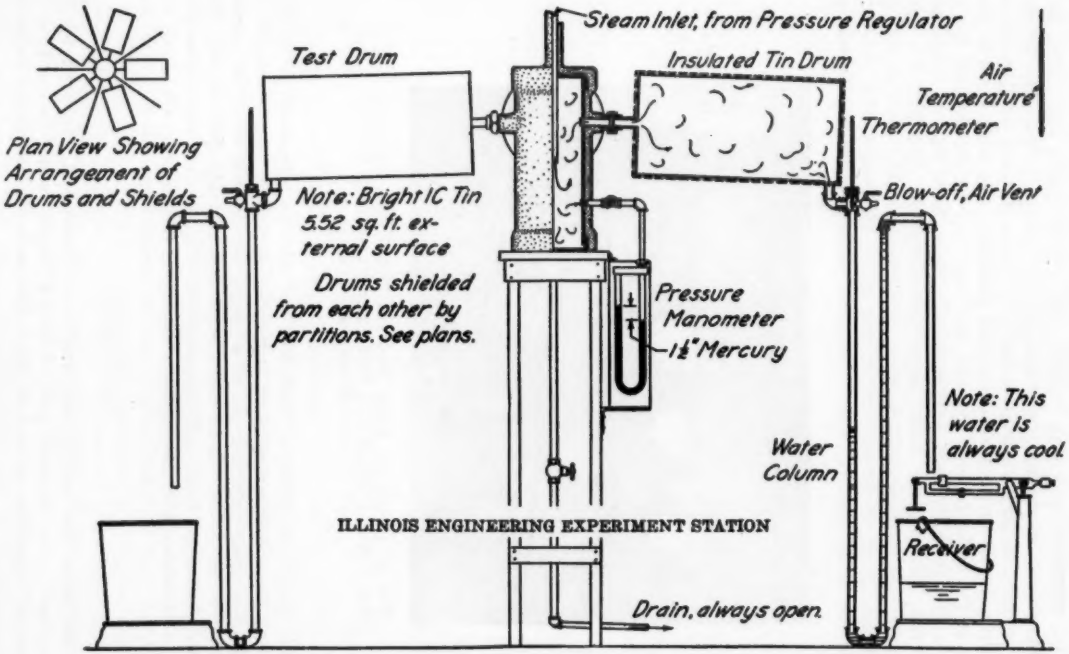
Steam Inlet, from Pressure Regulator
Test Drum
Insulated Tin Drum
Air Temperature Thermometer
Plan View Showing Arrangement of Drums and Shields
Note: Bright IC Tin 5.52 sq. ft. external surface
Blow-off, Air Vent
Drums shielded from each other by partitions. See plans.
Pressure Manometer
1½" Mercury
Note: This water is always cool
Water Column
ILLINOIS ENGINEERING EXPERIMENT STATION
Receiver
Drain, always open

FIG. 1. SECTIONAL ELEVATION OF STEAM DRUM PLANT FOR TESTING SURFACES FOR EMISSIVITY.
(AS USED IN UNIV. OF ILL. ENGRG. EXP. STA. BULL. 117)

on a horizontal diameter, where the metal was 7/32 or 0.219 in. thick. The average thickness of the cylinder was ¼ or 0.25 in.

The test procedure was very simple. All cylinders and seals were thoroughly blown out by raising the steam pressure just a trifle. The pressure was then reduced to the operating condition of 1½ in. mercury and the running seals allowed to fill up to normal running position. See Fig. 1. The thermometer at the outlet of each cylinder was read just before the test started, and the weight on each scale was recorded. Since there was no measurable pressure drop between the steam supply drum and the cylinders, the outlet thermometers always indicated the steam temperature at the start of the test, and continued to do so throughout the test *unless* air accumulated above the *water column* in the running seals. Frequent readings of these thermometers

FIG. 2. PLANT FOR HEAT TRANSMISSION TESTS, STEAM TO AIR, USING CYLINDERS OF DIFFERENT METALS AND SURFACE FINISHES

were made during the period of each test, and all tests terminated at the first sign of any drop in temperature at any outlet, at which time the weight on each scale was again recorded. In most cases, this did not occur for periods of from five to ten hours. In all cases, tests were repeated and checked. The condensate discharged from each running seal (Fig. 1) was taken through a short piece of rubber tube submerged in the bucket receiver on each scale.

TEST RESULTS

From the latent heat of the saturated steam condensed, corrected for 2.5 ft. of exposed ¼ in. diameter pipe, it was a simple matter to compute the heat given off from the surface of the cylinders by means of the following formula:

$$K = \frac{rW}{A\,(t_s - t_a)}$$

where

K = Btu per square foot per hour per one degree Fahrenheit
r = latent heat of one pound of the steam, in Btu
W = corrected weight of condensate per hour, in pounds
t_s = temperature of the steam, in degrees Fahrenheit
t_a = temperature of the surrounding air, in degrees Fahrenheit
A = area of the drum exposed to the air, in square feet

It should be noted that tests of the quality of the steam, with drain or bleeder at bottom of steam drum open, invariably showed better than 98 per cent and often better than 99 per cent. The amount of the ¼ in. pipe exposure correction was ascertained from calibration tests with all the ¼ in. pipe heavily insulated with hair felt.

The test results are shown in Table 1. Two values from previous work (University of Illinois) are included for comparison.

TABLE 1. RESULTS OF HEAT EMISSION TESTS WITH METAL CYLINDERS [a] CONTAINING STEAM AT ABOUT 215 F AND STANDING IN STILL AIR AT ABOUT 75 F

SPECIAL NOTE: In no case was the temperature of the outside surface of the metal as much as 0.5 deg. less than the steam temperature in the cylinder.

Kind and Thickness of Metal and Condition of Surface	Heat Transmission Coefficient K in Btu per Sq Ft per Hour per 1 Deg F
Cast iron, 0.25 in., foundry finish	2.21
Cast iron, 0.25 in., painted a dull flat gray	2.43
Sheet iron, No. 28 gage (0.0156 in.), very rusty	2.37[b]
Sheet copper, No. 28 gage (0.01264 in.), slightly dull, but not corroded	1.56
Sheet copper, No. 28 gage (0.01264 in.), recleaned like new	1.39
Sheet copper, No. 28 gage (0.01264 in.), clean new surface	1.41[b]
Sheet copper painted with blue-drab gray auto-engine enamel, 45 per cent pigment, 55 per cent varnish	2.42[b]

[a] Each cylinder was 10 in. diameter x 20 in. long, having an outside surface area of about 5.5 sq. ft.
[b] From previous work, University of Illinois.

DISCUSSION OF RESULTS

It is most essential to keep certain temperatures in mind in any discussion of the results shown in Table 1. The steam temperature in the cylinders was 215 F, ± 1 deg (F); the temperature of the inside surface of the cylinders against which the steam was condensing is not known, but it was almost identical with that of the steam; the temperature of the outside surface of the cylinders was well within 0.5 deg (F) of the steam temperature, probably very much less than 0.5 deg (F) in the case of the copper cylinders; and the temperature of the surrounding air and "visible" objects was approximately 75 F, ± 5 deg (F). Actual temperature measurements with the copper-constantan thermocouple at the air surface of the cast-iron cylinder ranged from 214.70 F to 214.60 F for the plain foundry finish and flat gray painted surfaces respectively, when the steam temperature was 215 F.

Since the resistance to heat flow into or out of a surface, or through a material from face to face, varies almost directly with the temperature differences, it will be apparent that the resistance at the outside surface of all the cylinders is, comparatively speaking, simply enormous. The temperature drop

from the outside surface *to the surrounding* "air" and "visible" objects is of the magnitude of $(215 - 75) = 140$ F, whereas the total temperature drop at the inner surface, and through the metal wall of the cylinders, is less, much less, than 0.5 deg (F). It will be further apparent that considerable changes in the thickness of the metal cylinder walls will have very little effect on the outer surface temperature and hence on the resistance (always nearly proportional in any given case to the temperature difference across the surface) at the air surface.

The "neck of the bottle" in all the cases shown in Table 1 is encountered as the heat attempts to pass from the outer metal surface of the cylinders to the surrounding "air" and "visible" objects. Just how this heat escapes from this surface depends on the well known laws of radiation, conduction and convection and is discussed in a subsequent section of the paper, but some reference must be made to the matter of radiation effects in discussing certain other aspects of the results in Table 1.

It has been shown that the metal thickness and conductivity are relatively unimportant when the cylinders are surrounded with air, but why should the thick, low-conductivity cast-iron cylinders lose *more heat* than the thin high-conductivity copper cylinders? The outer surfaces of all cylinders are at about the same temperature, but the *character of the surfaces* are quite different, and this condition has a material effect on the amount of heat given off from the surfaces by radiation. All bright, hard, mirror-like surfaces give off less radiant heat than dull, porous, non-mirror like surfaces *at the same temperature* and with the same surroundings. It has just been pointed out that all of these surfaces are within 0.5 deg (F) of the steam temperature, hence changes in *surface finish* may become very important, and it is found that, *regardless of the thickness or conductivity of the cylinders used in these tests,* the heat transmission (Table 1) varies with the surface finish, being a minimum ($K = 1.39$ and 1.41) for the new or recleaned copper cylinders, and a maximum ($K = 2.43$) for the dull flat gray painted cast-iron cylinder, while for a cast-iron cylinder with a natural foundry-finish $K = 2.21$. Even a slightly dull surface on a copper cylinder increased the heat transmission to $K = 1.58$. By merely painting the copper cylinder, its heat transmission may be still further increased to $K = 2.42$, obtained in earlier tests with this same plant. It is interesting to note that the painted copper cylinder transmits almost precisely the same amount of heat as the painted cast-iron cylinder. The K values are 2.42 and 2.43 for painted copper and painted cast iron respectively. Surface finish becomes a matter of prime importance whenever radiation plays an important part in the emission of heat from a metallic surface, as in these tests.

Finally, any heat which was not discharged or emitted from the cylinders by radiation must be removed by conduction and convection to the surrounding air and was doubtless more or less uniform in amount, since the surfaces of all the cylinders were at about the same temperature, and the cylinders were surrounded with air of uniform temperature having no initial motion. A detailed analysis of the problem in which a mathematical computation of proportional amounts of heat discharged or emitted from the cylinders by radiation, conduction and convection follows.

ANALYTICAL STUDY

A vast amount of experimental data have been collected by investigators working in the field of heat transfer by radiation, conduction, and convection, and involving plane, cylindrical, and spherical surfaces of various natures. These investigations have covered the range from wires only a few thousandths of an inch, to pipes several feet in diameter, and have included both natural and forced convection in various fluid media. More recently the attention of physicists has been directed to correlating this mass of data, making use of the laws of dimensional homogeneity, and the present theory has been perfected to such an extent that the heat emission from various surfaces by radiation and natural convection can be calculated with sufficient accuracy for all practical purposes, provided the shape can be evaluated in geometrical terms and the nature of the surface can be definitely specified. Hence, no further tests seem necessary to establish fundamental principles and constants. Most of the uncertainties arise in the cases of random shapes and of materials having surfaces of variable or indefinite texture, such that they cannot be identified with any of the physical constants now available. Tests on these special cases may be necessary in order to establish performance data for individual pieces of apparatus, but the results will be applicable only to that particular apparatus under the conditions existing at the time of the test. The tests reported in this paper serve very well to illustrate the degree to which the general theory can be made to apply.

In the case of the cylinders under consideration, the heat from the steam is transferred through the steam film on the inside surface of the cylinder and through the metal walls by conduction. It is then lost from the outside surface of the cylinder by radiation and conduction through the air film on the outside. The latter heat is finally removed from the air film by natural convection. Since the heat transferred through the steam film and walls by conduction is equal to that lost from the surface by radiation and finally by convection, the problem may be completely solved by evaluating the radiation and convection.

The radiation may be calculated from the well known Stefan-Boltzman equation

$$H_r = 0.1723\, e \left[\left(\frac{T_1}{100}\right)^4 - \left(\frac{T_2}{100}\right)^4 \right] \qquad (1)$$

where

H_r = heat loss by radiation, Btu per square foot per hour.
e = emissivity coefficient, decimal fraction of black body radiation.
T_1 = absolute temperature ($t_1 + 460$) of the hot surface, degrees Fahrenheit.
T_2 = absolute temperature ($t_2 + 460$) of the surrounding air and "visible" objects, degrees Fahrenheit.

Table 2 gives the values of e to be used in equation (1) for a number of materials in common use.

The convection may be calculated from Heilman's [5] modification of the Rice [6] equation.

[5] Surface Heat Transmission, R. H. Heilman (Mechanical Engineering, May, 1929, Section 1).
[6] Chester W. Rice (A. I. E. E. Transactions, Vol. 43, 1924).

TABLE 2. EMISSION COEFFICIENT (e) FOR VARIOUS SURFACES AND SURFACE TEMPERATURES

Surface	e_{100}	e_{200}	e_{392}	Source
Cast iron, oxidized	....	0.940	....	Hütte [a]
Rough steel plate	0.945	0.950	0.961	Heilman [b]
Asbestos paper	0.930	0.934	0.943	Heilman [b]
Flat black lacquer	0.960	0.980	....	Heilman [b]
Black lacquer	0.800	0.950	....	Heilman [b]
White lacquer	0.800	0.950	....	Heilman [b]
Copper, slightly polished	....	0.167	....	Hütte [a]
Copper, calorized	....	....	0.180	International Critical Tables
Copper, oxidized	....	....	0.568	International Critical Tables

[a] Des Ingenieurs Taschenbuch, 22nd edition.
[b] Surface Heat Transmission (Mechanical Engineering, May, 1929, Section 1).

$$H_c = C\left(\frac{1}{D}\right)^{0.2} \times \left(\frac{1}{T_a}\right)^{0.181} \times \Delta t^{1.266} \qquad (2)$$

where

H_c = heat loss by convection, Btu per square foot per hour.
C = a constant depending on configuration.
D = diameter, inches.
T_a = average of the *absolute* temperatures of the surface and the surrounding air, degrees Fahrenheit.
Δt = temperature difference between the surface and the surrounding air, degrees Fahrenheit.

Table 3 gives the values of the constants (C) to be used in equation (2) for various shapes of surfaces.

TABLE 3. CONFIGURATION CONSTANTS (C)

Horizontal cylinders	1.016
Long vertical cylinders	1.235
Vertical plates	1.394
Horizontal plates facing upward	1.790
Horizontal plates facing downward	0.890
Spheres	1.820

It may be noted that both equations (1) and (2) are based on the actual surface temperature. If only the temperatures of the steam in the cylinder and surrounding air are known, and the surface temperature and rate of heat transmission are unknown, it will be necessary to assume a reasonable rate of heat transmission and make a trial calculation for the temperature, and from this calculate the transmission. These results may then be used in a second calculation to obtain a closer approximation. Usually the second is sufficient. In the case of the cast-iron cylinder under consideration the tests had shown the value of K, or the heat emitted per square foot of surface per degree difference in temperature per hour, to be approximately 2.4 Btu and one trial calculation proved to be enough.

McAdams,[7] Sherwood, and Turner found the resistance of the steam film on the steam side to be $\frac{1}{2{,}000}$. The thickness of the wall of the cast-iron cylinder used for the tests was 0.219 in. at the point where the surface thermocouple was located. The conductivity of cast iron is given[8] as 325 Btu per square foot per degree difference in temperature per inch per hour. Hence, the resistance of the wall was $\frac{0.219}{325}$, and the total resistance of wall and film $= \frac{1}{2{,}000} + \frac{0.219}{325} = 0.001175$. The combined conductance would, therefore, be $\frac{1}{0.001175} = 850$.

Since the temperature of the steam in the cylinder was 215 F, the air temperature 75 F, and the coefficient of heat transmission was approximately 2.4, the probable surface temperature can be calculated from the equation

$$2.4\,(215 - 75) = 850\,(215 - t)$$
$$\therefore t = 214.6 \text{ F.}$$

This calculation indicates that the drop in temperature through the film and wall did not exceed 0.4 deg (F), and the surface temperature could be taken as 215 F with sufficient accuracy for all practical purposes.

The average of the surface temperatures, as observed by means of the surface thermocouple, was 214.70 F for the cast-iron cylinder with the foundry finish and 214.61 for the same cylinder painted flat gray. This is in remarkably close agreement with the calculated surface temperature of 214.6 F. Since the drop in temperature through the steam film and 0.219 in. wall of the cast-iron cylinder was less than 0.4 deg (F), it may be reasonably assumed that the drop in the case of the thin walled copper cylinder was virtually nothing.

Since all of the cylinders had the same outside dimensions, the heat loss by convection would be the same for all. This loss by convection may be calculated from equation (2). On cylinders of this size no appreciable error will be made by regarding the ends also as cylindrical surface, in which case the value 1.016 may be used for C.

$$H_c = 1.016\left(\frac{1}{10}\right)^{0.3} \times \left(\frac{1}{605}\right)^{0.181} \times (140)^{1.266}$$
$$= \frac{1.016 \times 530}{1.585 \times 3.19} = 107 \text{ Btu per square foot per hour.}$$

The heat loss by radiation may be calculated from equation (1). In the case of the cast-iron cylinder with foundry finish it is

$$H_r = 0.1723 \times 0.94\,[(6.75)^4 - (5.35)^4] = 205 \text{ Btu per square foot per hour.}$$

The total heat loss is, therefore, $107 + 205 = 312$ Btu per square foot per hour, and since the temperature difference from the surface to the air is 140 F,

[7] Heat Transmission in Surface Condensers, by W. H. McAdams, T. K. Sherwood and R. L. Turner (A. S. M. E. Transactions, Vol. 48, 1926, p. 1246).
[8] International Critical Tables, Vol. 5, p. 221.

$$K = \frac{312}{140} = 2.23 \text{ Btu per square foot per degree Fahrenheit per hour.}$$

This is in very close agreement with the value 2.21 determined by test for the same cylinder. The heat loss by radiation and convection, and values of K have been calculated in the same manner for other cylinders and are given in Table 4.

From Table 4 it may be observed that reasonably close agreement was ob-

TABLE 4. CALCULATED HEAT LOSS BY RADIATION AND CONVECTION

Cyl. No.	Kind and Thickness of Metal and Condition of Surface	Heat Loss, Btu per Sq Ft per Hour			K		
		Convection	Radiation	Total	By Calculation	By Test	Per Cent Diff.
1	Cast iron, 0.25 in., foundry finish.	107	205	312	2.23	2.21	1.0
2	Cast iron, 0.25 in., painted dull flat gray	107	214	321	2.30	2.43	6.0
3	Sheet iron, No. 28 gage, very rusty	107	205	312	2.23	2.37	6.0
4	Sheet copper, No. 28 gage, slightly dull, not corroded	107	124	231	1.65	1.58	4.0
5	Sheet copper, No. 28 gage, recleaned like new	107	37	144	1.03	1.39	35.0
6	Sheet copper, No. 28 gage, clean new surface	107	37	144	1.03	1.41	37.0
7	Sheet copper, No. 28 gage, painted with blue-drab gray auto-engine enamel, 45 per cent pigment, 55 per cent varnish	107	214	321	2.30	2.42	5.0

tained between the calculated and observed values of K in all except cylinders Nos. 5 and 6. In these two cases, the value $e = 0.167$ for slightly polished copper was used in the radiation equation (1). This value was probably obtained for a copper surface which had been machined and buffed. The surface of the rolled commercial sheet used in the test cylinder evidently did not correspond to this, but was oxidized to a certain extent. After the cylinder had been allowed to stand in the air for several months the surface was well oxidized, but not corroded. Hence, close agreement was obtained in the case of cylinder No. 4 for which the value of $e = 0.568$ for oxidized copper was used. The latter will probably be applicable for heat transfer apparatus that has been in use for any length of time. For new apparatus a value $e = 0.41$ seems to be indicated from the tests.

From Table 4 it is also evident that the variations in K, or the heat transmission from the cylinders, was brought about by variations in the radiation. The latter varied from 37 to 214 Btu per square foot per hour, or from 35 to 200 per cent of the heat loss by convection. Since radiation is purely a surface effect, and the drop in temperature through metal walls, even of considerable thickness, is negligible, it is also evident that the heat loss from metal cylinders with steam on one side and air on the other will be largely determined by the nature of the surface and not by the character or thickness of the metal

forming the walls. If the nature of the surface can be identified with any of the known physical constants for radiation and the shape can be evaluated in geometrical terms, it is possible to predict the heat emission very closely from the theory available.

It is also of interest to note, both from Table 2 and Table 4, that since radiation plays a very important part in the heat transfer under the conditions discussed, the various paints will have practically the same effect irrespective of color, provided these paints consist of a pigment with an oil, varnish, or lacquer base.

CONCLUSIONS

1. The conductivity of a metal may bear little or no relation to the amount of heat emitted from a cylinder made of that metal when filled with saturated steam and surrounded with still air.

2. The actual metal thickness, within rather wide limits, has little or no relation to the amount of heat emitted from a metal cylinder filled with saturated steam and surrounded with still air.

3. The surface finish is an all important factor in the heat emission from metal cylinders filled with saturated steam and surrounded with still air.

4. The resistance to heat flow from the outer surface of a metal cylinder filled with saturated steam, when still air is the surrounding medium, is relatively enormous when measured in terms of the resistance of the metal wall itself, or the resistance at the inner (steam) surface of the cylinder.

5. The heat flow from the surface of a metal cylinder filled with saturated steam, when still air is the surrounding medium, may be calculated with a degree of accuracy sufficient *for all practical purposes,* provided the nature of the surface can be definitely specified and conforms with the radiation constants now available.

DISCUSSION

F. C. HOUGHTEN: The results reported in this paper fill a great need judging from the inquiries that have come to the Society's Research Laboratory regarding the relative heat emission from various surfaces. These inquiries come not only from people interested in the use of radiation but also from manufacturers of radiation and they include direct radiators, concealed convectors and blast heaters. Other inquiries have come from manufacturers of pipe. As you probably know, the copper producers are at the present time looking with considerable interest to the use of copper pipe in steam and hot water heating systems and the use of tubing with thinner walls.

One of the factors of special interest is the relative heat emission from copper and iron pipe. The authors have pointed out that the heat emission can be divided into radiation and convection and, of course, the radiation losses depend chiefly upon the surface area and are not affected by other factors. The convection losses, however, are influenced by the shape; the character of the surface in the particular case of cylinders probably being affected by the diameters of the cylinder and whether or not that cylinder is horizontal or vertical.

Can the authors give any results from their data or any estimate from their experience concerning the relative heat emission from a small cylinder of the magnitude of pipe, particularly 1¼-in. pipe and under, and relative data on heat emission when those smaller cylinders are horizontal or vertical?

A. C. WILLARD: We have made no tests on smaller or larger cylinders. Professor Kratz undoubtedly will have something to say on the application of the theory to the smaller and also vertical cylinders.

A. P. KRATZ: The theory covers that point very well. I have made no calculations so I cannot give you any definite idea of the difference, but Heilman and also Rice, give a series of factors that can be used for cylinders in various positions, also for flat surfaces, and the equations take into account the diameter as well as the relative lengths.

L. W. SCHAD: The amount of heat dissipated by convection per unit area of surface increases with a decrease in cylinder diameter, while the amount of heat dissipated by radiation per unit area of surface remains the same. Hence the difference between the total heat dissipation from a cast iron cylinder and a copper cylinder would decrease with a smaller diameter.

Have the authors tried metallic paints or surfacing for the purpose of reducing radiation from cast-iron cylinders?

PROFESSOR WILLARD: Yes, we have taken the rusty sheet-iron drum or sheet-steel drum and painted it with aluminum paint and a material reduction in the heat emission resulted. This paper probably has no application to thin pipe radiators, pipes or tubes. It applies only to cylinders.

P. NICHOLLS: The authors state that their only object in making these tests was to convince the majority of users of some of the fundamental principles which they summarize at the end. I think, however, since they attempt to analyze their results they may regret that their experiments were not conducted with that exactitude that they would have desired if they had originally thought of analyzing them, because many readers will undoubtedly consult this paper for technical information probably not warranted by the type of experiments.

I question whether they are justified in taking the convection as they compute it as a proven figure. Such computed convection from the more or less irregular and not very long cylinder that they used might be liable to considerable error. They deduce—and do not in their argument question the accuracy of their deduction—that the radiation from the polished copper surface is much greater than would be expected. I compute roughly their emissivity as 0.55 instead of 0.167 as given in Table 2 for the slightly polished surface. That makes the radiation three times as big as the stated value.

As far as the object of the paper is concerned, that does not matter, but at present there is much interest—as evidenced by a later paper—on the effectiveness of air spaces where surfaces of a low emissivity are used. Many investigators are interested now in what are the theoretical and also what will be the practical emissivity of surfaces so employed. Will surfaces so used retain in practice the original bright condition in which they may be found in the initial tests?

The authors therefore say that the best they could do in polishing a copper surface was to get 0.55 emissivity for it instead of 0.167 which it should be. Assuming that they tried to polish them well, which can be done, I would think from their conclusions that the actual emission of 0.55 should be left open, and that it would require more accurate tests than possible with the type of apparatus they used to make it an assured fact.

PROFESSOR WILLARD: What Mr. Nicholls says is true, but I wish to emphasize that these tests were made on practical cast-iron surfaces as received from the foundry; that the copper surfaces used were commercial copper surfaces as received, whether cleaned by us or submitted by the manufacturer of copper sheets. Whether we could have obtained a higher polish of the surfaces is debatable. Mr. Nicholls indicates that we could. Had we done so the heat emission from the copper surfaces

would have been much less than it was, as the tests are reported and the difference between cast iron and the copper would have been much greater.

We realized that these tests are not of a high degree of scientific merit in the sense of refinements such as the length or diameter of the cylinders. The idea was to place before the manufacturers and the users of cast iron and other metallic tubes and cylinders, the basic conception that wall thickness does not necessarily indicate the relative amount of heat that will be transmitted, but that within certain limits the surface finish does determine how much heat will be transmitted. While our tests are based entirely upon work with some drums that are 10 in. in diameter and 20 in. long, the test results under the conditions of the test are defensible. I hope the Society's Research Laboratory will go further with this problem and will change these diameters and these lengths. I suggest that the cast iron, copper, steel, aluminum, brass, silver and anything that may be used be polished. This is merely an experimental fragment, gentlemen, thrown out to see if we can impress upon the industry the importance of knowing more about what determines the heat transmission and emission from commercial metals today instead of simply using our so-called common sense in making the comparisons.

Mr. Houghten has indicated that already the Laboratory is prepared to do much more, and much more should be done, but the tests as made represent exactly what happens with certain metal thickness and character of surface.

PRESIDENT HARDING: It appears to me that this paper is of sufficient importance to those who are interested in the manufacture of direct radiation to have more discussion.

R. V. FROST: The results of this work are of interest to manufacturers of radiation, particularly those making the fin type of radiator. In some work that has been carried on in the Frost Laboratory during the last couple of years to determine what metal has the greatest commercial value in the construction of a certain fin type radiator, we made comparisons of hard and soft copper, hard and soft brass, aluminum and different gages of steel and found that, from a commercial standpoint, steel was much ahead of other metals. This may be rather surprising because we so frequently hear the remark, "Why do you not make this radiator of copper?" If the radiator were made of copper, it would cost 4 times as much and would gain only 80 per cent in heat transmission, thus, its value in copper would be only about 50 per cent, that of steel, on a dollars and cents basis.

While copper has a heat transmission of 8 times that of steel, the rate of heat transfer from the copper or steel to the air is so small in comparison to the conduction through the metal itself, that nothing is gained. Therefore, steel is worth much consideration in the manufacture of fin type radiators.

W. H. CARRIER: Mr. Frost's remarks have quite an important economic aspect. There is one qualification which he might have included, as it is impossible to compare two radiators with different rates of transmission (I am speaking of forced convection radiators) on the basis of their transmission under a given air-flow condition alone. This point was thoroughly covered in a paper by Mr. Stacey who, I believe, is present. The relationship is not simple. It is conceivable that in a given amount and type of surface a good result could be obtained so far as the cost of that surface was concerned per square foot, but to obtain the necessary velocity flow over that surface the pressure might be doubled. When the pressure is doubled on one type of heater, it might just as well be doubled on the other and when this has been done much higher velocities have been obtained with no increase in power and a different relationship may exist on that basis.

In other words, the resistance of the heater must be taken as a factor in a definite mathematical way which has been set for it, or its equivalent, as well as the transmission under certain conditions.

Many papers have been presented at this Society in which that factor was either neglected or inadequately cared for, and yet I am concerned about the factor of resistance which is just as much a determining factor as the area of the surface or its cost per square foot.

R. C. BOLSINGER: The discussion seems to be getting away from the paper which gives much valuable data. Some people might obtain an erroneous idea from this paper about the radiation from cast iron. I do not think that cast iron would affect radiation to any great extent—at least not to the extent that some people might expect. The actual heating effect would depend entirely upon the design of the cast-iron radiator, including the element, if it was of the tube type rather than the slab type, and also upon the shadow effect which more or less would influence the radiation.

PROFESSOR WILLARD: I wish to endorse Mr. Bolsinger's remarks. The paper applies only to cylinders. It does not apply to commercial types of cast-iron radiators except to this extent: We have also taken commercial cast-iron radiators, accepted them as they came from the foundry and painted them with an ordinary oil finish, such as is used by the radiator manufacturers, and then in addition to that we have painted them with aluminum bronze.

In all cases we get the same effect that the paper has shown. When we apply a surface finish, which is like aluminum bronze, we cut down the heat emission. That reduction is largely as a result of reduced radiation, but the shadow effect Mr. Bolsinger points out, does cut in so that instead of getting a reduction of, say 50 per cent, we may get a reduction, depending on the type of radiator, of 10 or 15 per cent.

PRESIDENT HARDING: Briefly summarized, if all radiators had exactly the same external form and shape, no matter of what material they were constructed, and if they were all dipped in the same kind of paint, the answer would be the same as far as the heat emission is concerned. Am I right?

PROFESSOR WILLARD: If the radiators were all cylinders, I would agree with you.

No. 880

HEATING EFFECT OF RADIATORS

By R. V. FROST,[1] NORRISTOWN, PA.

MEMBER

IN A PAPER on enclosed radiation [2] presented before the Society at Boston in January, 1925, the writer submitted data on some tests conducted in 1914, showing that by merely placing a straight box enclosure about a cast-iron radiator [3] the average temperature at the breathing line of the room could be increased while at the same time the condensation from the radiator decreased. This, it is believed, was the first presentation of what is now known as *radiator heating effect.* The generally accepted explanation of this phenomenon is that the cabinet radiator distributes the heat at a lower level than does the direct radiator, but in tests of cabinet-enclosed, fin-type radiation conducted during the summer of 1930, the writer obtained data that give support to another explanation. This explanation, however, is not new, for the writer first suggested it in articles published in 1926, but he had at that time no way of definitely proving the theory involved.

These tests were conducted in a calorimeter room which, although similar in its general details to many other radiator test rooms in the country, differs from most of them in that the cooling effect is obtained by a direct method. Some experimenters have described it as a water-cooled wall method. It provides the following three methods of cooling:

1. Direct cooling by a sheet of cold water flowing over the outer wall surface, shown in the elevation, Fig. 1.

2. Cooling by air over the outer wall surface.

3. A combination of (1) and (2) with added cooling effect by evaporation.

The choice of the method depends upon the weather conditions and the conditions under which it is desired to test. In winter, air is used and in summer, water or air and water combined.

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[1] President, Frost Research Laboratory.

[2] *Some Facts About Enclosed Radiation,* by R. V. Frost (A. S. H. V. E. TRANSACTIONS, Vol. 31, 1925).

[3] The word *radiator* is used in the commercial sense and not in the scientific sense in this paper.

Presented at the 37th Annual Meeting of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, Pittsburgh, Pa., January, 1931.

DESCRIPTION OF TEST ROOMS

The general layout and dimensions of the rooms are shown by the floor diagram, Fig. 1. The outer walls and the floor and ceiling of the calorimeter room are insulated with 12 in. of cork, thus making possible the maintenance of extremely uniform conditions. This uniformity has been demonstrated, when, with a wide variation in the outside temperature, the temperature within the test room did not vary one-half degree over a 24-hr period.

The calorimeter room contains two separate test rooms of equal dimensions, which are neither communicating nor have they a common wall except

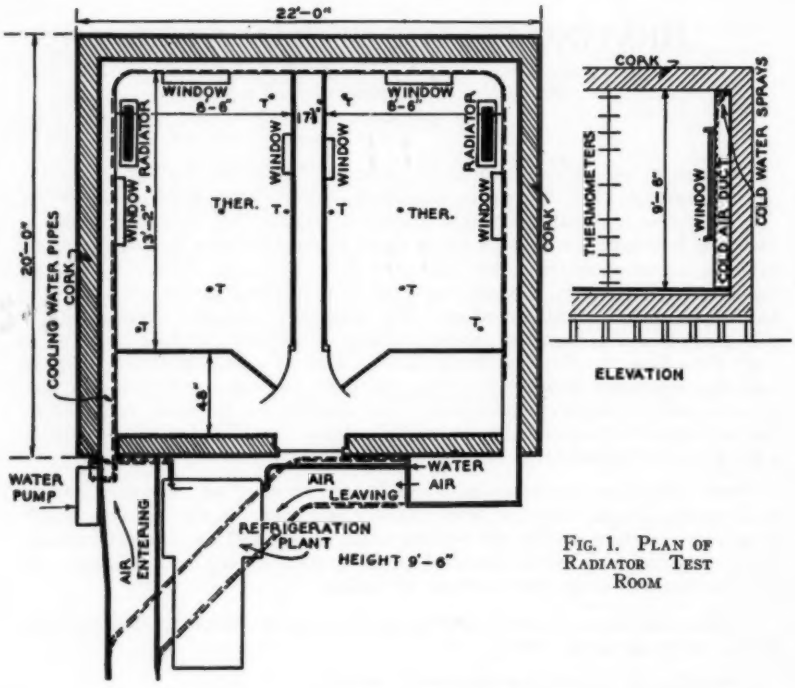

FIG. 1. PLAN OF RADIATOR TEST ROOM

through a connecting passage. The cooling surface walls are of sheet metal painted with light yellow semi-gloss oil paint. They comprise walls on two sides, or one-half the total wall surface of each test room and each wall contains one double-hung window fitted with metal weatherstrips, or two windows per room.

When air is used for cooling, the velocity over the wall surface is equivalent to a 12-mile wind, while the wind pressure, when infiltration is considered, can be made equivalent to the pressure of a 40-mile wind. When water is used for cooling, the water for each room is drawn from the same cooling

tank but passes through separate meters, valves and pipes so that it is possible accurately to balance the cooling effect on the two rooms. This arrangement has made it possible to obtain exact determinations of heating effect. It was found that a variation of one-half degree in room temperatures upset the

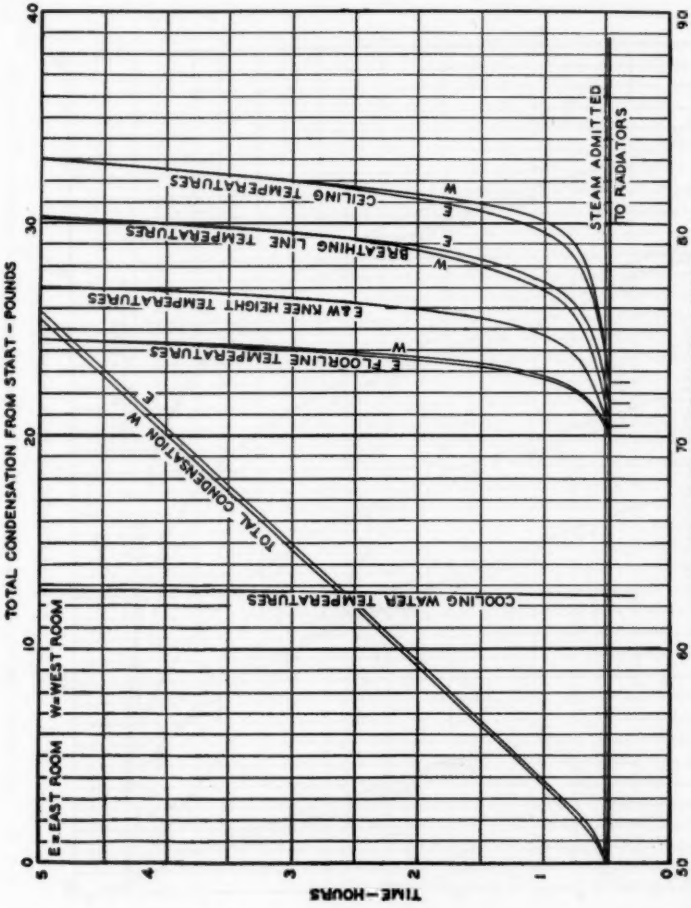

FIG. 2. COMPARISON OF RADIATOR TEST ROOMS WITH IDENTICAL RADIATORS

conditions quite seriously and it was therefore necessary to secure a very exact balance between the two rooms to be certain of the results.

PROCEDURE FOR MAKING TESTS

The procedure in these tests was first to balance the two rooms by testing two radiators of identical size, one in each room, and adjusting the cooling

water flow over the walls of the two rooms until the temperature conditions in the two rooms were exactly the same for the same rate of condensation. Such a balanced condition is shown in Fig. 2.

Radiator *A* was then placed in one room and radiator *B* in the other. The

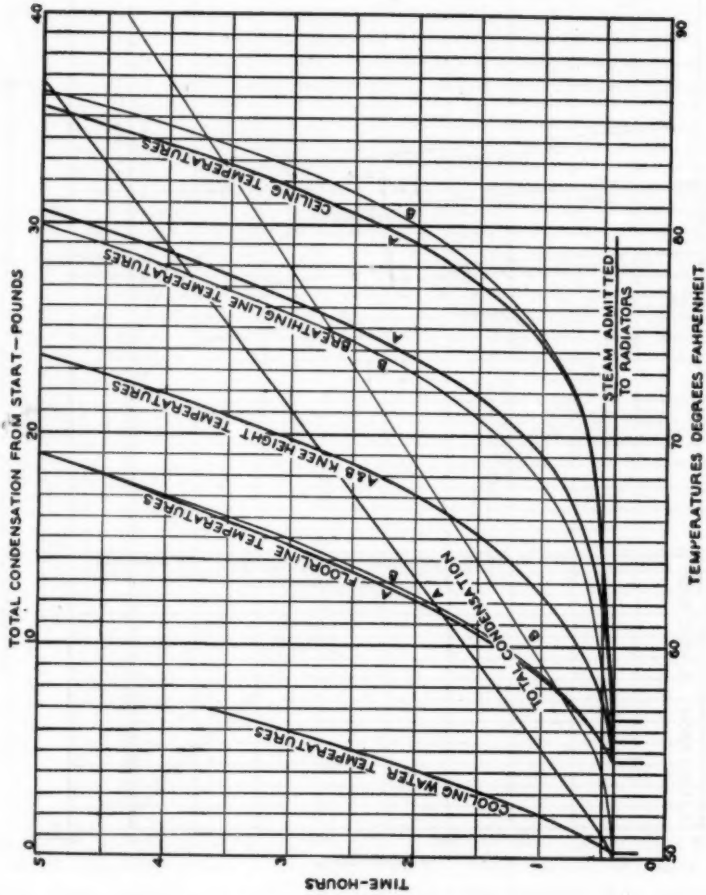

FIG. 3. COMPARISON BETWEEN FIN-TYPE (*A*) AND CAST-IRON (*B*) RADIATORS (TEST 1)

locations of the radiators are shown on the plan of the test room (Fig. 1) and it will be noted that they are in relatively the same location in each room. After the radiators had each been tested in one room they were interchanged and the tests repeated. The results of these tests are shown on Figs. 3 and 4. Both room temperatures and condensation are shown on these charts. The left

ordinate represents time, the lower abscissa is room temperature, and the top abscissa, condensation.

The condensation as plotted represents total condensation from the start of the test. Four temperature lines are plotted, namely, average temperature of

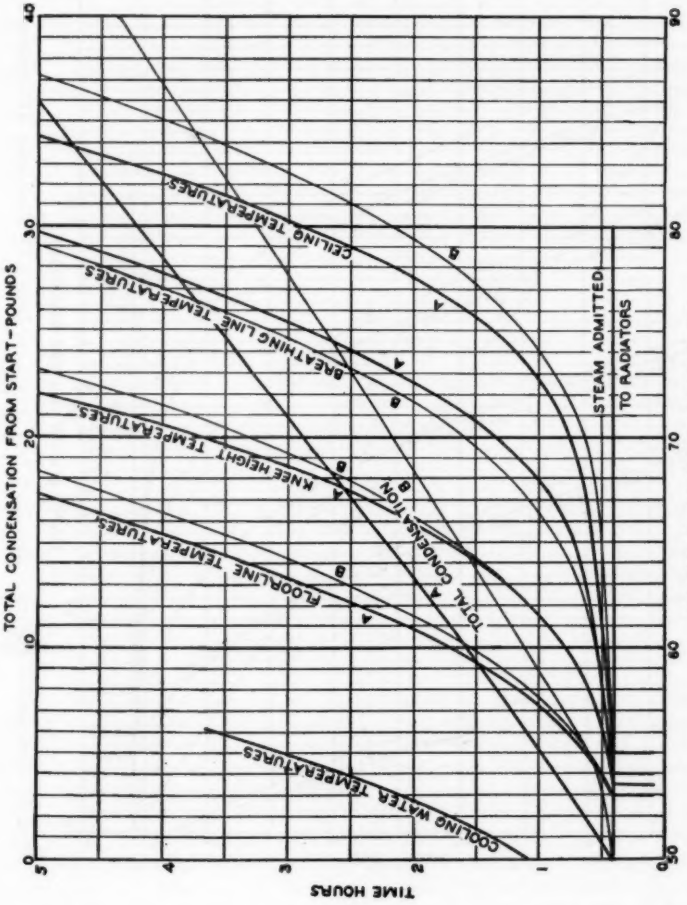

FIG. 4. COMPARISON BETWEEN FIN-TYPE (*A*) AND CAST-IRON (*B*) RADIATORS (TEST 2)

the room on the levels of 3 in. above floor, 20 in. above floor, 60 in. above floor and 3 in. below ceiling. In addition to these, temperatures at 10 in., 30 in., 40 in., 80 in., 90 in. and 100 in. were also recorded and are indicated on Fig. 6. Condensation and room temperature for both rooms are plotted on the same chart to give a graphic comparison.

It will be noted that there is some variation between Figs. 3 and 4, although the two rooms had been previously balanced very closely. This variation was due largely to the wide calibration of the thermometers. To compensate for

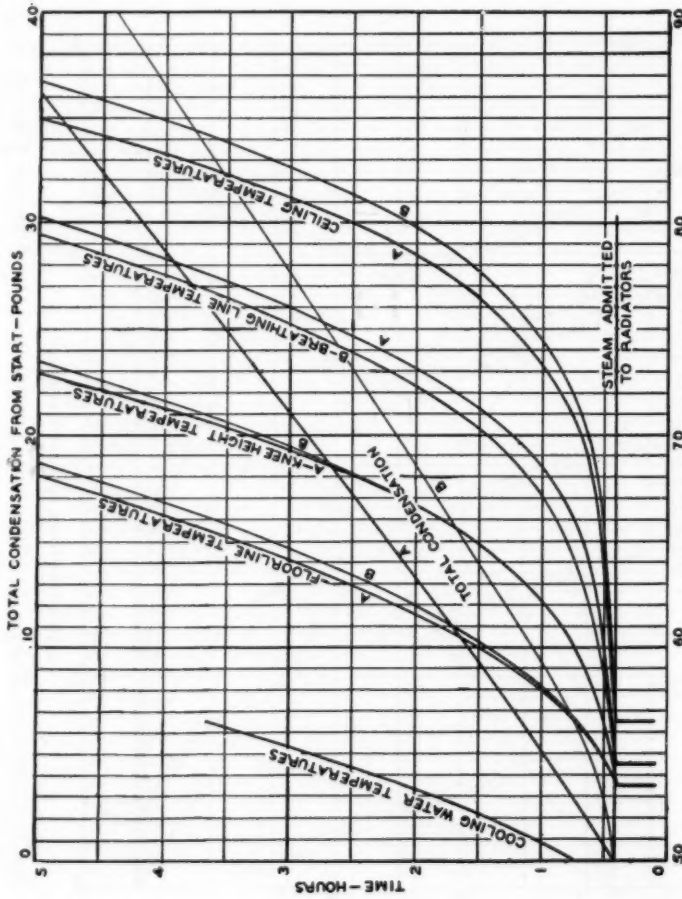

FIG. 5. COMPARISON BETWEEN FIN-TYPE (*A*) AND CAST-IRON (*B*) RADIATORS (COMPOSITE OF TEST 1 AND TEST 2)

this variation, Fig. 5 was constructed and this is regarded as the official chart of the test. It is a composite, average chart of Figs. 3 and 4.

One objective was to have the floor line and knee height temperatures of the two rooms coincide. This was accomplished as closely as possible by proper selection of the radiators used for comparison. By having this condition, it was possible to assume that the temperature conditions surrounding

the two radiators were identical and so to offset the possibility of the claim being made that the test conditions were unequal.

In the study of the curves, consider first the temperature lines. The short vertical lines at about 55 F at the bottom of Figs. 3, 4 and 5 represent the temperatures in the rooms before the start of test. Steam was then admitted to the two radiators simultaneously as indicated by the horizontal lines, and the temperatures in the two rooms began to rise immediately. It should be noted that there is a certain definite relationship between the four levels plotted and that after the first quick rise in temperature the lines flatten out to a slightly curved gradual increase that becomes less curved as the temperatures approach a higher level. In other words, the temperature lines take an elliptic shape that eventually leads into a vertical line as the heat exchange becomes perfectly balanced.

It should be noted that the ceiling temperature for Radiator *A* was much lower than for Radiator *B*, but that the breathing line temperature for Radiator *A* was higher than for Radiator *B*. In other words, the difference between the ceiling temperature and the breathing line temperature for one radiator is much less than for the other.

ANALYSIS OF RESULTS

For the purpose of comparison, the cooling water temperature is shown in its relationship to the room temperatures. It bears a very close relationship, at least in appearance, to floor line temperature.

The condensation lines are very slightly curved as is to be expected due to the fall in condensation with a rising room temperature. When a balanced condition is reached without a rise in temperature then the condensation lines will become straight lines. An interesting observation is to be noted in comparing the lower sections of these condensation lines. One represents the condensation from a fin-type radiator having but a proportionately small weight, and the other is from a cast-iron radiator. The effect of the additional weight in the cast-iron radiator is shown by the immediate increase in condensation from this radiator. It should be noted that after the cast-iron radiator is heated up the condensation line begins to straighten out.

The important feature about the condensation from the two radiators is shown in the gradual spreading of the two condensation lines. It is apparent that the fin radiator equipped with a cabinet heats the zone from the floor to the breathing line with a much lower hourly rate of condensation. On this chart (Fig. 5) a graphic explanation of the entire phenomenon of heating effect is given. Two radiators used for heating two rooms of equal cooling effect, start with the same room temperature at the beginning of the tests, and build up the temperatures of the two rooms over a period of several hours at precisely the same rate in the zone occupied by the radiators, that is from floor to knee height, each radiator operating under the same temperature conditions, and yet the one with the cabinet condenses less steam consistently throughout the period.

It would appear that the explanation lies in the lower temperature at the ceiling, but the curves of Fig. 6 might indicate otherwise. These curves are for the average temperatures from floor to ceiling for each radiator. One

set of curves is plotted on the basis of a temperature of 70 F at the knee height level and the second at a temperature of 70 F at breathing line level. The averages for the curves are tabulated on Fig. 6. It will be observed that for a temperature of 70 F at the knee height level, the average temperatures for the full height of the two rooms are but 0.1 deg apart and likewise when the averages are taken for 70 F at the breathing line the difference is 0.1 deg. It is true that the averages for the zone from floor to breathing line are higher for the cabinet radiator than for the exposed cast-iron radiator but with each room raised to the same average temperature over the entire room it is apparent that each room has absorbed virtually the same amount of heat from

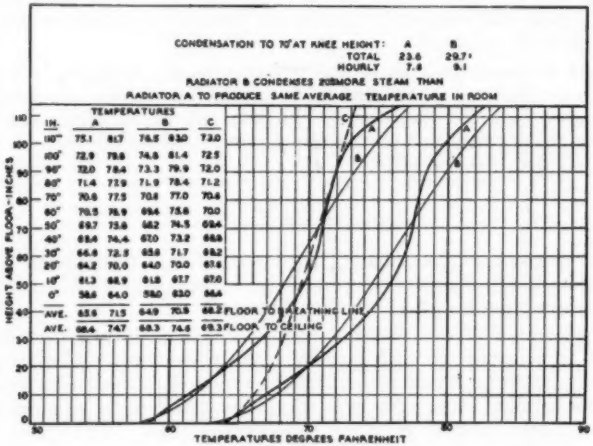

FIG. 6. TEST ROOM TEMPERATURES—FLOOR TO CEILING

each radiator. Therefore, the cause for the difference in condensation rate must lie in some other explanation.

For the usual type of room construction and with a 20-deg difference between the wall temperature and the average room temperature, the hourly rate of condensation for this particular type of cabinet was about 20 per cent less than the hourly rate from the exposed radiator. But if a shield is placed in back of the exposed radiator, the rate of condensation from this radiator now decreases so that the percentage in favor of the cabinet drops to about 14 per cent.

EXPLANATION OF RESULT OBTAINED

This indicates that the cabinet prevents the dissipation of heat by radiation, a large part of which passes out through the windows and walls of the room. By placing a shield between the radiator and wall a portion of the radiation loss is prevented and if the room were completely insulated to prevent the

loss of all radiant heat so that the heat from the radiator were used only to heat the air in the room the heating effect of the exposed radiator would then be equal to that of the cabinet radiator.

Rooms are built with considerable glass surface and as the ratio of glass to exposed wall surface in recently constructed buildings is on the increase rather than decrease, an increasing radiation loss from this source is probable. Therefore, in the design of a radiator it is well, if fuel economy is a factor, to give consideration to the loss of radiant heat and to design the radiator to give the highest heating-effect ratio. In fact, it is evident that heating effect, in comparing the value of radiators, will become of increasing importance.

Returning to a discussion of the test room, it is apparent that the wall temperature is indicated by the temperature of the water flowing over it, since the wall material is sheet steel, painted with a yellow gloss oil paint. This material having a high conductivity, transmits the heat to the water with a minimum resistance, so that although there is some loss, it is negligible and cannot be detected in the water temperature.

It has been claimed that this wall construction is not comparable to normal heating conditions. The only difference between this wall construction and the usual wall type is that this wall responds to temperature changes quicker than would a standard exterior wall, and that under the method of testing used, a condition was duplicated in a few hours that required days to work out in the usual problem of heating. Fig. 7 shows the conditions for normal heat transmission compared to the test room used. In zero weather the inside wall temperature is approximately 58 F, the temperature gradient being as shown on the left. To obtain a wall temperature of 58 F in this test room, the cooling water was maintained at 58 F, but as far as the surface with which the air of the room was in contact or which received the radiant waves of heat, there was no difference between the standard wall and this test room wall, provided both walls were painted surfaces of the same color. The wall in these test rooms will not transmit heat any more rapidly under given inside wall and air temperature conditions than will the standard wall at zero outside. In fact, these tests have shown that for the total area of exposed wall surface in the test rooms the heat demand is about equivalent to that of the same area of standard wall in zero weather.

There are many kinds of wall construction, all with different rates of heat transmission and a radiator test room built up of thick walls similar to those of a building, could not be used to study the effect of different rates of heat transmission. The metal, water-cooled walls, on the other hand, have the advantage of flexibility, and can by proper regulation of the cooling water temperature be made to duplicate almost any type of wall construction. This is illustrated in Fig. 7 in which a standard type of building wall is shown on the left, with the temperature gradient from 70 F inside temperature to zero outside, shown dotted. On the right is shown the metal, water-cooled wall of the test room. As the metal wall in contact with the cooling water assumes the temperature of the cooling water within a small fraction of a degree, it is apparent that cooling water at 58 F will duplicate zero degrees outside in the particular wall shown on the left. Other walls can be similarly simulated; for example, 50 F cooling water would correspond to an 8-in. brick wall,

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unfinished inside, or a 16-in. stone wall (sandstone or limestone), unfinished inside.

The test rooms were not built so much to comply with standard test conditions as to provide a means of studying heating effect. The tests reviewed in this paper were among the first conducted specifically in this laboratory for that purpose, and the results have been sufficiently encouraging to warrant the belief that it will be possible to gather further data on the subject of heating effect that will clear up many other questions on the matter of radiator heating.

CONCLUSIONS

In arriving at conclusions regarding the value of heating effect, the effect due to radiation loss should not be confused with the improved air distribution

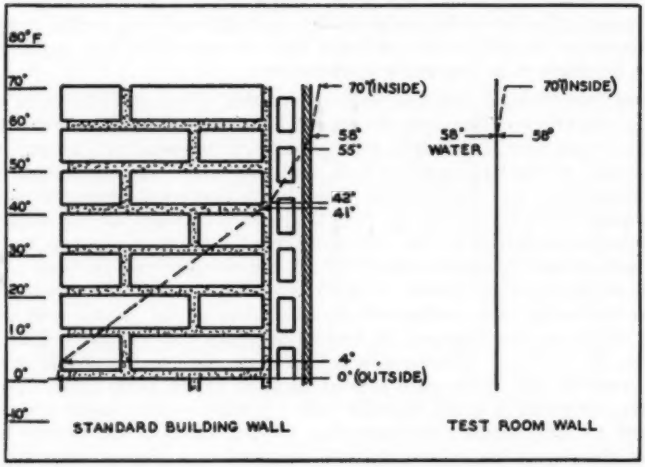


FIG. 7. TEMPERATURE GRADIENTS THROUGH STANDARD WALL AND TEST ROOM WALL

in the room. The tests demonstrated that not only does the cabinet over the radiator prevent the loss of a large percentage of radiant heat, but that it also distributes the heat at a lower level.

The two factors cannot be determined by the same tests. These tests were intended to prove that there is a loss of radiant heat and to determine the percentage of that loss. They have also shown that there is a better distribution of the heat when a cabinet is used, but the value that can be placed upon that better distribution, this study does not indicate.

It is permissible to assume, however, from the data here developed, that a cabinet over a radiator can save at least 20 per cent by better utilization of radiant heat, and an additional percentage of a definite value, by better distribution of heat in the living zone. To determine the exact percentage of heat saving due to better distribution requires a different method of test than

the one described in this paper and this percentage is dependent upon many factors in addition to the cabinet.

It would be very desirable to combine these two factors in order to determine the full value of heating effect, for it is apparent that many of the new types of radiators cannot be installed on the old condensing basis of rating. One such radiator by the mere diffusion of heat at a low level has been found to heat the zone from the floor to 3 ft above the floor to the same temperature as another that condenses 60 per cent more steam.

With this type of radiator the temperature at the breathing line will be lower than with another but to a person sitting in the room the degree of comfort with the first is higher than with the second for the reason that the major portion of the body is in a higher temperature, as indicated by line *C*, Fig. 6. If this radiator is installed on the basis of its condensing capacity, overheating results. Therefore, the condensing capacity of this radiator is not an accurate index of its useful heating capacity.

The desirability of a new method of rating radiation is apparent. For years the assumption has been accepted that the relative humidity is an index of heating comfort. Is it not possible that uniformity of temperature, not on a time scale, but uniformity on a scale of vertical height, is an equally important index of comfort? Certainly if one's integument must adjust itself to changes of temperature of from 5 to 10 deg every time a change is made from a standing to a sitting position, one's constitution will be subjected to a severe strain.

The problem of radiator heating does not, in the writer's opinion, involve merely the emission of heat. The matter of heating effect is equally as important.

DISCUSSION

PROF. A. C. WILLARD, PROF. A. P. KRATZ AND M. K. FAHNESTOCK (WRITTEN): This paper emphasizes the distinction between the *heating effect* and the *steam condensation* of radiators. An interesting plant has been used for showing this distinction and making striking comparisons between the performance of different types of radiators in respect to *heating effect* and *condensation*.

The plant is quite unique and no doubt could be operated to simulate actual normal heating service conditions, but it was not so operated in the tests presented in the paper. Instead, only one condition of heating service was provided for in making these tests, and that was the *warming up* condition from the state of cold rooms, walls, and radiators. No data are presented on the *cooling down* of the rooms, walls, and radiators which would have been decidedly more favorable to the radiator which showed up badly under *warming up* conditions. This radiator would have also performed much better if tested under *normal heating service* conditions with the radiator, the room, and the outside in thermal equilibrium, a condition which was never realized in any of these tests.

Specifically, we wish to point out that:

(1) The inner metal wall surface temperatures were not constant nor uniform, since the cooling water temperatures varied from 50 to above 56 F (Fig. 5), and the paper states "the metal wall assumes the temperature of the cooling water within a small fraction of a degree."

(2) Such low wall temperatures are possible in actual heating practice at zero degrees but not usual, and naturally direct *unenclosed* radiators will show up to

a greater disadvantage in such an extreme low temperature wall environment than an enclosed radiator.

(3) As an illustration of this, it may be noted that at the end of the first ½ hr (starting with air at knee height at 54.5 F and walls at 50 F as shown in Fig. 5) the unenclosed radiator had condensed $\frac{4.5}{3.5}$ lb or twice as much steam as the enclosed radiator. However, if we compare the *last* ½ hr of available data we find the ratio becomes about $\frac{9}{4.5} = 1.29$, or only 29 per cent greater, and the room is still far from being in a state of equilibrium.

(4) It is also rather surprising to discover that the condensation rate comparison between the two types of radiators on an average hourly basis (Fig. 6) shows neither the true instantaneous condensation rate at the time the room attained 70 F at the knee height nor the true average rate based on the cumulative condensation at this same time. The condensation rate as computed is a fictitious rate corresponding to the slope of a line connecting the point of zero condensation or starting point of the test with a point representing the amount of condensation accumulated at the time the room attained 70 F.

(5) The paper points out that the metal wall construction responds quickly to temperature changes. Therefore, the room should reach an equilibrium condition quickly if the wall surface temperature is held constant, as shown in Fig. 2. In the tests reported in Figs. 3, 4, and 5, however, this characteristic of the room construction was not realized because the wall temperature was not held constant. Therefore, the air in the rooms and the room walls did not attain an equilibrium condition even after 4½ hr, as is shown by the rising temperature curves of Figs. 3, 4 and 5.

(6) The two tests, Nos. 1 and 2 (Figs. 3 and 4) do not check each other, and the explanation that this is "due to the wide calibration of the thermometers" does not seem adequate. The calibration should have taken care of any differences in thermometer readings resulting from errors in the thermometers. The true explanation is that the rooms were never operated under equilibrium conditions and hence it was impossible to duplicate conditions on two successive tests. The magnitude of this variation between tests Nos. 1 and 2 is as great as that of the variation (Fig. 5) between the two types of radiators on which the conclusions are based.

(7) In studying *heating effect* with a given outside temperature or weather condition, the inside wall surface temperature must be a definite function of the radiator being tested, that is, any given radiator should and will in practice establish its own inside wall surface temperature.

The fallacy of testing with walls at a fixed or predetermined temperature, rather than permitting the radiator to establish this temperature is clearly shown when it is pointed out that a shield back of the exposed radiator reduced the condensation 6 per cent. In an actual building, the inside surface temperature of the wall back of an exposed radiator would be something over 100 F instead of ± 58 F, and any shielding effect would probably be less than 6 per cent. A similar condition, but of different magnitude, exists for the other walls in the room as well as for the wall back of the radiator.

There is nothing more confusing in reporting research work than the presentation of partial or incomplete results, no matter how interesting such results may be, and it is hoped Mr. Frost will "carry on" with his research work in this field and give us complete results from his plant, including "cooling down" and "normal heating service" conditions.

E. G. SMITH (WRITTEN): In the conclusions of this paper the author makes the following sweeping claim: "It is permissible to assume however, from the data here developed, that a cabinet over a radiator can save at least 20 per cent by better utilization of radiant heat and an additional percentage of definite value by better distribution of the heat in the living zone." Before this statement is accepted, the basis upon which it rests should be carefully analyzed.

Let us assume for the present that the statement is true and then let us see to what it leads. On the standard building wall (Fig. 7) a line is shown which represents the temperature gradient through the construction and the probable temperatures at various points are indicated. Assuming that the conditions represented by this figure could be produced by a cabinet radiator, let it be further assumed that the cabinet radiator is removed and replaced by an exposed radiator which will still maintain an inside air temperature of 70 F, but will emit 20 per cent more energy in so doing. Since the wall is opaque the only way that 20 per cent more heat can get through it is for the temperature gradient through the wall to be increased by 20 per cent. In the figure the inside temperature of the wall is 58 F and the outside temperature is 4 F, a difference of 54 F. If we increase 54 F by 20 per cent we get 68.8 F which must be the temperature drop through the wall if the rate of heat loss is to be increased by 20 per cent. Since the temperature of the outside surface of the wall must remain about 4 F, it follows that the temperature of the inside of the wall would have to be 4 F plus 68.8 F or 72.8 F. Then the temperature of the inside of the wall would have to be higher than the temperature of the air in the room, so that instead of absorbing heat from the air in the room the wall would give off heat to the air and the radiator would not only be unable to emit any heat by convection but would actually have to absorb a little.

It might be argued that all this radiant energy went out through the windows but this too is impossible because, to the long heat waves emitted by a steam or hot water radiator, a window pane is practically as opaque as a sheet of metal. Furthermore if the radiator is placed under the window, as it should be, the glass will be largely shielded from the direct radiation by the sill, and finally, glass has a smooth surface which is a good reflector and as a result it does not absorb as much radiance as would a rougher surface such as paint or plaster. A glass surface does let out a great deal of heat but it does so by the conduction of the heat itself not by the transmission of radiant energy.

Having proved that a 20 per cent difference in efficiency would not be possible for the standard wall (Fig. 7) it remains for us to show why it was possible for the water-cooled wall shown in the same figure. In order to understand clearly why this can be possible, it will first be necessary to consider what happens as a result of radiant energy striking the inside of the standard wall. The first effect is to raise the temperature of the inner surface of the wall and thereby to slightly increase the rate of heat transfer through it, but unless the radiation is intense this increase is small. The second effect is that as a result of the increased temperature of the wall surface the rate of heat loss from the air in the room to the wall is reduced. Thus a radiator that emits a large amount of radiant energy will be required to emit less heat by convection than would be the case if it emitted little or no heat by radiation.

On the other hand, it is not possible for the radiance that strikes the water-cooled wall to raise the temperature of the inner surface of that wall appreciably. Instead, the flow of water over its outer surface holds the temperature of the wall down to practically the same figure that it would have for a shielded radiator. Under such circumstances it is not surprising that a large portion of the radiance emitted by the radiator is lost.

One of the most convincing ways to see what misleading results can be obtained by means of a water-cooled wall is to calculate the probable temperature drop through such a wall. Referring again to Fig. 7 in the paper, let us assume that the

water-cooled wall at the right is transmitting 13 Btu per hour per square foot and further assume that it is $\frac{1}{32}$ in. thick, the temperature difference between the inside of this wall and the outside then comes out to be about $\frac{1}{1000}$ F. Thus if we assume Fig. 7 to be fairly representative we may expect the temperature drop through a standard wall to be about 58,000 times as great as through a water-cooled wall. It is this enormous difference in the temperature drop through the walls that made it possible to get results with a water-cooled wall that would have been impossible with a standard wall. To make the importance of this difference perfectly clear, it may be well to recall that at the beginning of this discussion it was pointed out that one of the important effects of radiant energy in a room is to raise the temperature of the inner surfaces of the walls and thereby reduce the rate at which the air in the room is cooled by contact with the walls. In the case of a standard wall an increase of several degrees in the temperature of the inside surface of the wall can be accomplished with an increase in temperature drop through the wall of only a few per cent and a correspondingly small increase in the rate of heat loss from the room. Furthermore, even this small percentage is offset by the presence of extra radiant energy in the room. On the other hand, to raise the temperature of the inside surface of a water-cooled wall by even 1 deg would require an output of radiant energy which might be equal to 1,000 times the entire convected output of the radiator. From these considerations it must be clear that practically all the radiant energy absorbed by a water-cooled wall is lost and of course it follows that a water-cooled wall cannot be used to compare the efficiencies of radiators.

Another difference between a room having water-cooled walls and an actual room in the winter time is, that in the case of an actual room the temperature of the inside surface of the windows is colder than any other surface in the room. This could not possibly be true of a room having water-cooled walls.

Still another difference lies in the temperature of the infiltered air.

It therefore seems logical to conclude that for the purpose of testing radiators the only satisfactory way to simulate an actual standard wall is to build the test wall with the same kinds of surfaces and the same over all thermal conductivity as the standard wall.

Another factor to be considered in running comparative tests on radiators is the type of thermometer bulb used. An entirely shielded bulb should, of course, never be used because it is not affected by the radiance or lack of radiance in the room. For the same reason a plain glass bulb should not be used because glass, especially when backed by mercury, is far too good a reflector. Thermometer bulbs used for this purpose should have an absorption coefficient comparable to that of a person's clothing. Because of the rough and porous surface of ordinary winter clothing it may be impossible to make a thermometer bulb as highly absorbent as ordinary clothing, but the dark gray paint used to finish beaver board for blackboards should give a fairly good approximation. The word rough as here used does not necessarily mean rough to touch. From the standpoint of its ability to absorb radiance velvet is one of the very roughest of surfaces because of its deep pile. For ordinary rooms the difference in readings obtained by blackened and unblackened bulbs will probably be small and may be negligible.

To return again to the subject of how heat leaves a room, there is one source of radiant energy loss that so far in this discussion has not been mentioned. The portion of a standard building wall immediately behind the radiator is likely to be raised to a temperature considerably above the average temperature of the air in the room. Probably it often would be good economy to back an exposed radiator by an insulating shield having a good reflecting surface such as high gloss baked enamel. Since the wall area involved is not great this loss is small except for poorly constructed walls, but even so the shield may often prove a good investment.

The case of the cabinet versus the exposed radiator boils down to the following 4 questions:

(1) Can an exposed radiator be made as beautiful as a cabinet?

(2) Can an exposed radiator produce as low a floor to ceiling temperature differential as a cabinet radiator?

(3) Can a cabinet radiator efficiently offset the discomfort caused by the loss of radiance to cold surfaces such as windows by the occupants of a room?

(4) Can the cost of a cabinet radiator be brought as low as the cost of an exposed radiator?

In considering the relative merits of cabinets and exposed radiators it may be well to mention a fact that frequently does not receive the consideration that it merits namely, that a hot water radiator, whether cabinet or exposed, can produce a more comfortable room than a corresponding steam radiator and often can do it with less fuel because the larger surface and lower temperature of the hot water radiator produces a lower floor to ceiling temperature differential than is possible with the smaller hotter steam radiator. This is especially true of open expansion tank systems.

Conclusions

(1) For the purpose of comparing the efficiencies of radiators the only allowable way to simulate a given wall is to build a test wall having the same types of surfaces and the same over all thermal conductivity. A water-cooled wall should not be used.

(2) The thermometers used in these tests should never be completely shielded and should preferably have bulbs with blackened surfaces so as to approximate as nearly as possible the high absorbing power of rough winter clothing.

(3) Whenever a cabinet radiator shows measurably greater economy of fuel than an exposed radiator, its superiority is probably due, mostly, to a lower floor to ceiling temperature differential rather than to a more efficient use of radiant energy, although in some cases, as an exposed radiator, an appreciable amount of radiance loss may be due to the overheating of the outside wall immediately in back of the radiator.

(4) Top floor rooms possibly may give different results from rooms on lower floors and corner rooms may give results different from rooms having only one exposed wall.

(5) Long low exposed radiators placed under windows tend to offset the loss of radiance by persons occupying the room to the cold surfaces of the windows, and if the radiance from such a radiator is well distributed and not too intense it may help to make the room more comfortable.

(6) One of the surest ways to produce a comfortable room is to insulate it thoroughly and thus increase the average temperatures of the inside surfaces of the walls. Double windows also make a room more comfortable.

(7) Hot water radiators may be expected to produce more comfortable rooms at a possibly lower fuel cost than steam radiators because of the lower floor to ceiling temperature differential. This is especially true of open expansion tank systems.

R. V. FROST: As to the cooling down of the rooms, we also worked out that phase of the problem, using a thermostatic control on the room, but no thermostat would give a really balanced condition. When we used a thermostat, it overemphasized the heating-up of the cast-iron radiation, the excessive condensation from the radiator building up the heat before the thermostat would act to shut off the steam. The result was a fluctuating temperature in the room. In the cabinet radiator, with its smaller steam volume, the thermostatic cycles were more rapid and the temperature line as a result was more nearly level.

As the normal heating condition we chose one of about zero outside temperature.

If we had taken a higher normal temperature, we could not have obtained the most severe conditions under which the radiator would be required to operate.

Professor Willard's criticism that our inner wall surface temperatures were not constant, nor uniform, since the cooling water temperature varied from 50 to 56 deg, can be answered by reference to Fig. 2 which shows the effect of normal wall temperature conditions, the effect upon the radiators being practically nil.

He points out that we had much higher condensation in the cast-iron radiator during the first ½-hr period. That is because of the accumulation of heat in the metal of the radiator. It is a practical demonstration of the heating-up factor, now considered so important in figuring the heating load upon a boiler. A cast-iron radiator has a much higher heating-up factor than the fin type cabinet radiator.

The fact that tests 1 and 2 do not check one another is better than to have them check exactly, because it shows the variation between maximum and minimum, and by taking the average between the two, we show the worst conditions under which the radiators can be compared. With this less advantageous condition, the fin radiator, in each case, shows a much better living zone temperature condition than does the cast-iron unit.

The most valuable criticism Professor Willard offers is the fact that cooling the walls by water gives us a uniform temperature of the wall from the floor to the ceiling, while under ordinary conditions, with air outside of the wall, we would have a higher temperature at the ceiling, than at the floor. This condition is covered in our test room equipment, by using air instead of water for cooling, but owing to the conditions under which we conducted these tests, with temperatures around 90 to 95 deg outside, we were obliged to use refrigerated water for cooling exclusively, instead of a combination of air and water. When the weather conditions are cooler we use either air alone or air and water. In severe winter weather we use air alone and then get conditions that are practically identical to those of a normal wall.

We grant that we are only showing partial or incomplete results in this paper, but owing to the breadth of the subject, it will take a long time and the work of several laboratories, before the value of heating effect can be properly demonstrated.

This, you may consider as a partial or a progress report and is of value only in that way. It is expected that we will eventually carry on other research which goes further into the subject of heating effect than we reported in this paper.

D. E. FRENCH: I would like to observe that this is a fine contribution which, however, is of the greatest value as a guide to the test work that must still be done. I hope that many laboratories will be added to those that we know already to be working on this subject. If we are to make the most value of the observations of this test and of the discussion that follows, we must not, however, make too broad constructions from the data that have been presented.

Mr. Frost has not described the exact radiators that were used. It must be recognized that the results of the paper apply accurately only to those radiators that he did use. Had the height of the cast-iron radiator been different, its heating effect undoubtedly would have been different. If the height or the type or the resistance of the thin type radiator been different, its effect might have been different. The results that Mr. Frost reports are similar to those obtained in other tests, but the influences of the various factors that are involved in the design of the various types of both cast-iron and fin type radiators have not been brought out clearly. It would be valuable if other laboratories worked on such phases of the problem as the following: (1) height of the radiator; (2) influence of the design of the fin type radiator on resistance; (3) amount of air that will pass through the radiator for a given heat transfer from the radiator to the air; (4) the effect of these several factors on the degree of stratification.

I question whether the author intended to make the statement, that if the room were completely insulated to prevent the loss of all radiant heat, so that the heat from the radiator were used only to heat the air in the room, the heating effect of the supposed radiator would then be equal to that of the cabinet radiator. If the heat by radiation were entirely eliminated there would still be a difference in the losses due to the difference in temperature between the floor and ceiling. If it were possible to heat the air in the room to a uniform temperature from floor to ceiling by varying the design of the radiator, the losses would be less than that of a radiator that allowed a differential temperature between the floor and ceiling.

E. K. CAMPBELL: I think Mr. French touched on a vital point in this discussion. The enclosed radiator converted what is wasted, in the unenclosed radiator in the form of radiant heat, into convected heat, thereby increasing the volume of air moved and lowering the temperature of that air. It is well known in warm air work that a larger volume of air at a lower temperature gives a lower differential between the floor and the ceiling and a more efficient result, particularly in large rooms. We must soon come to some system of measuring the volume of air moved by a given radiator and the corresponding temperature produced.

When Dean Anderson was in charge of the Laboratory, I asked him if there was any information available on the amount of air moved by a direct radiator but no such information was available then and there appears to be none now. As I see the situation set forth in this paper, the essential difference between the two radiators is in the volume and temperature of the air moved. You can speak of it as radiant loss or in any other way you please, but the problem is to save the radiant heat, convert it into convected heat, thereby increasing the volume of air and lowering the temperature.

R. C. BOLSINGER: One thing that was brought to my attention by this paper is the question of comparing the two types of radiators, which was briefly mentioned by Mr. French. I think that the cast-iron radiator, which was used in this test, was about the worst type that could have been used. A point that was stressed was the radiant effect because of the large glass surface used in buildings of today. In my opinion that would depend entirely upon the size of the room and what was placed in the room because these things would influence the heat loss by radiation through the glass.

I would suggest that Mr. Frost continue his experiments, using radiators much lower and narrower, where the flue effect would be considerably less and where the dissipation of heat waves would be greater. I think that the result would be lower temperatures between floor and ceiling.

Research engineers can clear up another point on which opinions differ and that is the correct height to take temperature measurements. Personally, I think the proper point is between 20 and 25 in. rather than 35 in. or the breathing line at 60 in.

W. H. CARRIER: I am greatly interested in this subject from the standpoint of what constitutes comfort in heated buildings. I believe that it is less important to save 20 per cent of our steam (although that is desirable) than it is to have a nearly uniform temperature from the floor to the 5-ft level. Many of us have been in rooms where our heads seemed to be at fever heat and yet our feet were cold. Such conditions are more often caused by poor building construction than by the heating system.

Our objective in heating is to produce comfort, not to obtain certain engineering results, but if both things can be accomplished economically, well and good. Mr. Frost's paper tends to show that it can be done, although I believe that he did show greater temperature variation in the lower levels with a heater that was most economical from the standpoint of steam consumption.

Temperature is important and it is involved in the test which we are discussing. The temperature which we should read is the temperature that the body feels, exclusive of the effect of moisture, and that can only be obtained with a blackened bulb thermometer. While the ordinary mercury thermometer indicates temperature variations, the sensation we feel of combined radiation and convection from our bodies is practically that of a blackened bulb thermometer. I believe a blackened bulb thermometer, which would receive radiant waves not only from the radiator itself, just as one's body does, but also from the cold surfaces of windows and walls, is essential in measuring room temperatures. If it is air temperatures that we are taking (and air temperatures are only part of the things that affect us), we may be entirely misled by the results.

The ideal system of heating, where uniform temperatures prevail in the living area, was described by Messrs. Campbell and French. If we should test a panel system of heating we might discover that the radiating effect from the ceiling and walls overcame the stratification of the air. Here again we could only obtain accurate results with a blackened bulb thermometer.

The work of Mr. Frost shows the possibilities of changing our method of heating and also shows the need of extensive study of certain phases.

E. R. QUEER: This paper is excellent in that it suggests the necessity for proper testing of radiators. There may be a misleading point brought out in this paper from the fact that radiation from radiators is not desirable for good temperature distribution.

Professor Willard and Mr. Fahnestock in *Heating, Piping and Air Conditioning*, March, 1930, show how increasing the heat radiated from a cast-iron wall type radiator gave a small temperature drop between the floor and ceiling. In one set of data reported, the radiated heat was increased and the size of the radiator was decreased from 36 to 21 sq ft. The temperature distribution was improved and the condensation reduced. The heat supplied to the room in the latter case was considerably less than in the former. However, the degree of comfort was increased materially. This was undoubtedly caused by better warming of the walls of the room, thereby decreasing body radiation. Here the radiant heat to the windows was increased and at the same time the heating effect was improved. In the work reported by these men the actual floor temperature, ceiling temperature and condensation were not given for the radiators so comparisons cannot be carried further.

MR. FROST: I appreciate the complimentary remarks that have been made, but I had hoped for more thunder.

As to what Mr. French had to say about the type of radiator, his remarks are pertinent but it is also true that every type of radiator has a different heating effect. I do not mean to say that you can take any two radiators, or the product of two different manufacturers, and secure different heating effect values, but there are now coming forward so many different types of radiators, that there is sure to be a wide spread in the value of what we call heat effect; that is, the distribution of temperature from floor to ceiling.

As to what Mr. Carrier had to say about comfort, comfort and economy seem to go hand in hand. A radiator that gives a better comfort effect in a room, a better temperature from floor to, say, 36 in. above the floor, is also the most economical type in the use of heat.

This subject necessarily must be carried on to much greater length than I have given here today and it is hoped that we can have a greater comparison on the subject than we have had heretofore. The work that Professor Willard and his associates are doing at the University of Illinois is probably the most forward-thinking work on the subject and there is close agreement in the result that the

two laboratories are obtaining in general, but he has not carried on to quite the same extent in the fin type radiator as we have.

The work that we did was for one manufacturer and necessarily we cannot report on the results of work on other types of radiators. For that reason I have not covered effective heating from a variety of types of radiators.

R. A. MILLER: In Mr. Frost's discussion of radiation and in that of some of the other men who brought up arguments at that time, the radiation through glass surfaces was mentioned and the statement was made that since rooms are built with considerable glass surface and as the ratio of glass to exposed wall surface in recently constructed buildings is on the increase rather than the decrease, an increasing radiation loss from this source is probable.

In view of known conditions where windows have been considered direct sources of heat loss, to make the statement that radiation losses through glass are practically nil will probably sound quite ridiculous. However, since we are considering heat losses at relatively low temperatures, in the infra red range of energy waves, comprised between approximately 1 millimicron or rather 1 micron, and 100, the amount of heat radiant energy loss through glass by direct radiation is quite important.

The actual area found under the heat curve for total radiation from a body at 126 F comprises some 259 sq C. The area which it is possible to transmit at 125 F through glass, or which it might be possible, comprises 1.44 sq C or 0.556 per cent of the total energy radiated by the body at 126 F.

Of this $\frac{5}{10}$ of 1 per cent, only 9.4 per cent is actually radiated, transmitted as radiant energy through the glass. This 9.4 per cent of the possible transmitted energy represents only 0.05 or $\frac{5}{100}$ of 1 per cent of the total available energy which is transmitted by a glass of a thickness of 2 mm.

In view of this it seems that in discussions by this Society of heat losses, radiation through glass should be rather warily handled. If you chose to go to transmission by means of conductance, I have no present figures available to refute the argument. Taking radiation by its usually accepted term I hope that I have entirely covered the argument of any radiation loss through glass.

We are planning to make some thorough investigations on heat transfer and heat losses through various glass and building panels, with a view to determining what particular advantages we may find from small air spaces contained between two glass plates. The work which has been done by Professor Willard on the transmission through thin shells of copper and steel with a highly reflecting surface would probably have some bearing on our problem. We appreciate the fact that glass is being more generally used in these new buildings. Double glazing is a likely development and it is one of our aims to assist the members of this Society and the allied organizations to understand the phenomena of glass transmission.

MR. CARRIER: If I understand the last speaker correctly, the objection to Mr. Frost's statement, if you made such, is that heat was transmitted through the glass by radiation. It would be rather interesting if you could give us some figures on the amount of reflection of heat.

MR. MILLER: At the moment I cannot give you any definite figures on the reflection of heat. On the direct reflection of visible light, the surfaces of plate glass reflect 4 per cent of the light coming to them, no matter what the angle may be. For the reflection of other radiating energy I am not at present prepared to state just what reflection will be.

MR. CARRIER: I take it that you would discount the loss by about one-half acting as a radiant shield.

MR. MILLER: I am not quite sure that I would like to even go that far at present. The loss through glass apparently is entirely of conductance. Where the heat on

both sides of the glass is within a relatively narrow range of temperature, there is little transfer of heat from one side to the other. However, where the glass is exposed to a low temperature on one side and a relatively high temperature on the other, there may be quite a transfer of heat by direct conductance not by radiation. Radiant energy, however, is entirely cut off. Even some of the visible rays in the red portion of the spectrum are completely cut off by the interposition of a plate of normal plate glass.

MR. CARRIER: I asked this question to clear up the matter. Will you make any statement that you have on this?

MR. FROST: This is a very interesting angle of the subject. As Mr. Carrier said, there is evidently radiation to the glass surface and then absorption of heat by the glass and then conductance from the outside surface of the glass to the outside air. It is a combination of radiation and conductance.

MR. MILLER: Shall we say absorption?

MR. FROST: It is a combination of radiation and conduction, because the conductance from the cool air of the room to the glass surface would be much less than radiation from a warm surface to the glass surface.

MR. MILLER: Mr. Carrier asked me about the reflection of the radiant heat. I said that I was not exactly prepared to state what that reflection would be. The reflection of visible light I know. At the present time we have no figures on the reflection of light below the visible spectrum.

In the question of absorption, the absorption in some glass is so great that the glass becomes hot enough to fracture and break. In clear white glass the absorption apparently is quite low.

The amount of radiant energy absorbed in normal room windows, etc., either is dissipated immediately by conductance through the rest of the glass surface to the outside air, or else most of it is reflected, because the temperature of the glass surface is relatively that of the outside air.

MR. FROST: There is a higher conductance, then, through the glass with a higher surface temperature on the inside?

MR. MILLER: Yes, the conductance of the glass is apparently the prime factor rather than the transmission of radiant energy directly. Direct radiation apparently goes through the glass in imperceptible amounts. In other words, you can cut off entirely all effect on a thermocouple by interposing between a red hot object and the thermocouple a single plate of plate glass 2 mm thick. There will be no heat or no radiant energy reaching the thermocouple.

MR. FROST: The difference that we tried to bring out in the paper was that the radiant energy from the warm surface of the radiation is dissipated through the glass surface much more rapidly than the heat transmitted by the air to the glass surface.

MR. MILLER: The main point that I wanted to bring out was the distinction between conductance and radiation.

MR. CARRIER: That is a good point. Two years ago I thought it was possible for heat to radiate directly through glass, but I found out that I was wrong in the low wave lengths. It might be assumed that heat will go through glass that you can see through.

Perhaps the statement you made should be qualified somewhat, simply in the interest of known facts. A radiator tends to radiate to the cold glass surface, which would tend to heat it up. In turn it would convect and also radiate to the outside. In other words, it is acting as a radiation shield.

MR. MILLER: It will do that. I expect that some tests will be made soon along this particular line and we are hoping to report some of the results to this Society.

No. 881

DEVELOPMENT OF THE VENTILATION SYSTEM OF THE HOLLAND TUNNEL

By A. C. DAVIS,[1] NEW YORK
MEMBER

THE Holland Tunnel between New York City and Jersey City, N. J., was the first ventilated vehicular tunnel started in the United States. Prior to starting the construction of this tunnel, the principal vehicular tunnels in operation were the Blackwall Tunnel under the Thames River, in London, England, completed in 1897; the Rotherhithe Tunnel, also under the Thames, London, completed in 1908; the Glasgow Harbor Tunnel, Glasgow, Scotland, completed in 1895; and the Elbe Tunnel, Hamburg, Germany, completed in 1910. None of these tunnels at the time of completion was equipped with mechanical ventilation. In 1925 a mechanical ventilating system was installed in the Rotherhithe Tunnel.

With no precedent as a guide, the study of the ventilation of The Holland Tunnel was begun in 1919. The following problems had to be solved:

1. Amount and composition of exhaust gases from automobile engines while operating on the highways.

2. Effect of such gases on animal life.

3. Dilution necessary in order that persons may breathe the exhaust gases for short periods of time without injurious effects.

4. System of ventilation best suited to vehicular tunnels.

5. Power required to ventilate the tunnel.

6. Demonstration of the operation of the system.

The best available data on the gases given off by automobile engines were those of Herbert Chase.[2] The tests on which these data were based were

[1] Assistant Superintendent, The Holland Tunnel.
[2] See *The Automobile*, by Herbert Chase (Vol. 30).
Presented at the 37th Annual Meeting of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, Pittsburgh, Pa., January, 1931.

conducted on the streets of New York, on varying grades, types of pavement and at various speeds. Owing to the limited number of types of automobiles tested and to the dissimilarity of grades and speeds as compared with those proposed for the tunnel, definite conclusions could not be based on these studies. For similar reasons, the results obtained by Alleman, Ballantyne, Watson or Hood and Kudlich could not be used.

EXHAUST GASES FROM AUTOMOBILE ENGINES

An agreement was entered into with the U. S. Bureau of Mines to perform the tests on the streets of Pittsburgh, Pa., having grades and pavement similar to those proposed for the tunnel. The selection of types of motor vehicles (see Table 1) for tests was based on traffic observations at the ferries across the Hudson River. All vehicles tested were those in actual use on the streets of Pittsburgh; no adjustments were made in carburetors and no change was made in the brand of gasoline used.

In general, the tests were made under the following conditions:

TABLE 1. CLASSES OF MOTOR VEHICLES TESTED

Class	Description	No. of Cars Tested
1	Five-passenger cars	19
2	Seven-passenger cars	13
3	Trucks up to 1½ tons carrying capacity	12
4	Trucks 1½ to 3 tons, carrying capacity	22
5	Trucks 3½ to 4 tons, carrying capacity	17
6	Trucks 5 tons and over, carrying capacity	18
	Total	101

1. Car standing, engine racing.
2. Car standing, engine idling.
3. Car accelerating from rest to 15 mph up 3 per cent grade.
4. Car running 3, 10 and 15 mph, up 3 per cent grade.
5. Car running 3, 10 and 15 mph down 3 per cent grade.
6. Car accelerating from rest to 15 mph on level grade.
7. Car running 3, 10 and 15 mph on level grade.

Fig. 1 shows the apparatus for collecting the exhaust gas sample from the exhaust manifold of an automobile. The samples were collected in sampling tubes and taken to the chemical laboratory for analysis by the modified Orsat method. Fig. 2 shows the testing apparatus installed on a truck. In the front is shown the apparatus for collecting the exhaust gas sample, and in the rear, the gasoline measuring apparatus is shown.

SUMMARY OF IMPORTANT RESULTS

From the standpoint of tunnel ventilation, the results of the road tests on 101 motor vehicles were briefly as follows:

1. Automobile exhaust gas consists of carbon dioxide (CO_2), carbon monoxide (CO), hydrogen (H_2), methane (CH_4), oxygen (O_2), nitrogen (N_2) and water vapor. The relative proportions of these constituents varies greatly

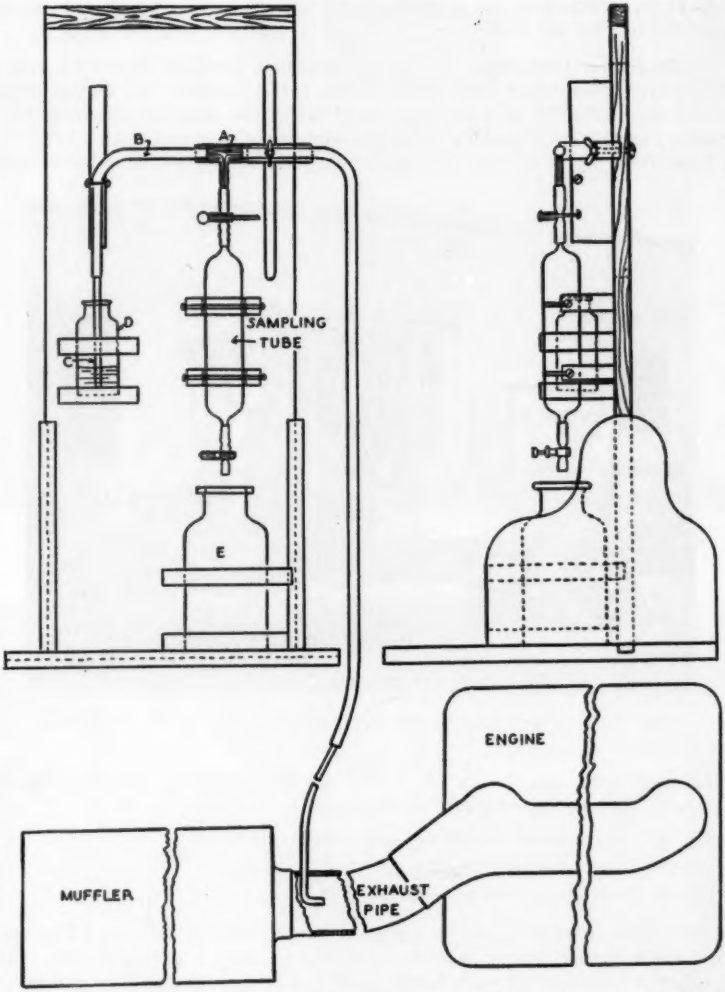


FIG. 1. APPARATUS FOR COLLECTING EXHAUST GAS SAMPLE FROM THE EXHAUST MANIFOLD OF AN AUTOMOBILE

in the exhaust from different motors, depending upon carbureter adjustment, degree of atomization, compression and other variable factors.

2. The important constituent of exhaust gas, as regards tunnel ventilation, is carbon monoxide (CO).

3. The CO found in the exhaust gas of the various industrial cars varied from 0.5 to 14.0 per cent.

4. The higher percentages of CO are produced when the throttle is nearly closed, as when running down grade or the car is standing and engine idling. The largest quantity of CO is produced when the gasoline consumption is greatest, as with cars accelerating or running upgrade at maximum speed.

From these tests and with the proposed number and types of vehicles using

FIG. 2. EXHAUST GAS TESTING APPARATUS INSTALLED ON A TRUCK

the tunnel at the specified speeds, it was calculated that the following quantities of CO would be produced in the tunnel:

On the up grades—0.118 cfm per foot of tunnel.
On the down grades—0.05 cfm per foot of tunnel.
On the level grades—0.09 cfm per foot of tunnel.

The motor vehicles to use the tunnel were classified as follows: Passenger cars, trucks under 2 tons capacity, 2 to 5 tons, and 5 tons and over. The ventilation requirements were based on 20 per cent passenger cars and 80 per cent trucks.

EFFECT OF AUTOMOBILE EXHAUST GASES ON ANIMAL LIFE

To determine the effect of automobile exhaust gases on animal life and to ascertain the dilution necessary to make such gases safe to breathe, investigations were carried on at the U. S. Bureau of Mines Experimental Station at *Yale University*. The experiments were performed in two types of chambers,

FIG. 3. GAS-TIGHT CHAMBER FOR DETERMINING EFFECT OF AUTOMOBILE EXHAUST GASES ON ANIMAL LIFE

FIG. 4. BRICK CHAMBER FOR DETERMINING EFFECT OF AUTOMOBILE EXHAUST GASES ON ANIMAL LIFE

one a gastight chamber having a capacity of 226 cu ft (Fig. 3), and the other a brick chamber having a capacity of 12,000 cu ft (Fig. 4).

Members of the investigating staff spent periods of one hour in the gastight chamber containing chemically prepared carbon monoxide varying from 2 to 10 parts per 10,000 parts of air. Groups of students, as well as members of the investigating staff, spent periods of one hour in the larger chamber in various concentrations of carbon monoxide, produced by the automobile shown in Fig. 4. The rear wheels of the automobile were removed and replaced by paddle-wheels to distribute the CO throughout the chamber. Fig. 4 also shows a meter for measuring the quantity of exhaust gas produced by the engine. Samples of the air in the chamber were taken periodically to determine the concentration. Subjects were examined by a doctor before and after each exposure, and samples of blood were taken and analyzed for CO content.

The conclusions drawn from these tests were, briefly, as follows:

1. Exhaust gas of automobiles contains no substance which is toxic to any appreciable extent, other than carbon monoxide.

2. Gasoline engines, with cylinders missing or when cold, may throw off disagreeable vapors irritating to the eyes and nauseating to some persons.

3. The physiological effects of carbon monoxide are wholly due to the union of this gas with the hemoglobin. To whatever extent the hemoglobin is so combined, by that amount it is rendered incapable of transporting oxygen to the organs and tissues of the body. The combination of CO and hemoglobin is a reversible reaction, so that when a person returns to fresh air the CO is gradually eliminated.

4. Of all physical signs and tests of CO poisoning, headache proved the most definite and reliable. Concentration of gas too weak or periods of exposure too short to induce this sign, are to be considered harmless.

DILUTION OF EXHAUST GASES NECESSARY

No one had this symptom (headache) to an appreciable degree after a period of one hour in the chamber with 4 parts of CO in 10,000 parts of air. With 6 parts the degree of effect, if any, was usually very slight, while with 8 parts there was decided discomfort for some hours, although not enough to interfere with the continuance of efficient work in the laboratory or at the desk. In any atmosphere containing a uniform amount of CO, almost all the gas inhaled at first is absorbed, but the successive increments of gas absorbed thereafter grow less and less as time goes on. If a man exercises sufficiently to double his volume of breathing, he absorbs as much CO in one-half hour as he does at rest in one hour.

Fig. 5 shows the rates of saturation of carbon monoxide hemoglobin and percentages of frontal headaches at various times of exposure when subjects are at rest, as when riding in an automobile. Based on the results of the investigations and on the estimated traffic capacity with assumed classification of traffic, it was determined that the following quantities of fresh air would be required per minute per foot of tunnel:

Down grade—125 cu ft
Level grade—225 cu ft
Up grade—295 cu ft

On this basis a total of 3,761,000 cu ft of fresh air per minute was required for the entire tunnel. The amount of air to be exhausted was calculated to be 4 per cent in excess of the fresh air supply.

SYSTEM OF VENTILATION BEST ADAPTED

Prior to starting these investigations, studies were made to determine the system of ventilation best adapted to vehicular tunnels. These studies, which involved 21 schemes, included longitudinal, transverse and combinations of these two methods of ventilation. As a result of these experiments it was

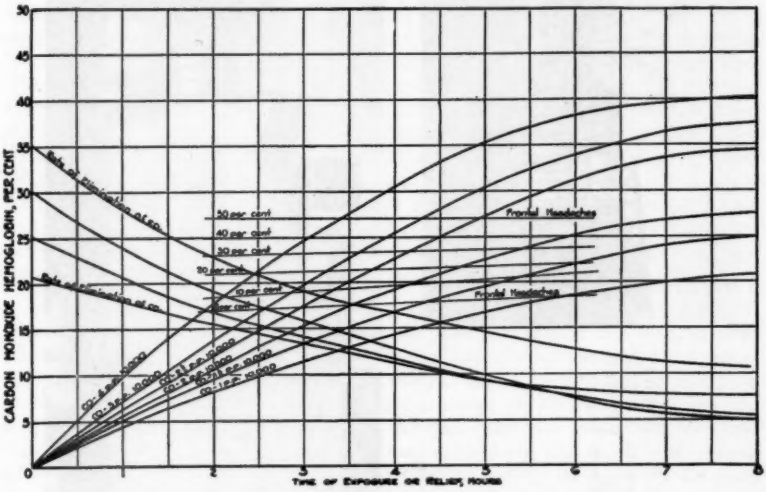


FIG. 5. RATES OF SATURATION AND ELIMINATION OF CO—HEMOGLOBIN AND PERCENTAGES OF FRONTAL HEADACHES AT VARIOUS TIMES OF EXPOSURE OR RELIEF, SUBJECTS AT REST

decided to use the transverse method of ventilation, but no definite decision was made as to whether the upper or lower duct should be used for the fresh air supply. The studies indicated that the lower duct should be used for the fresh air supply and on this basis experiments were made to determine the power required to ventilate the tunnel.

POWER REQUIRED TO VENTILATE THE TUNNEL

These experiments were conducted at the Engineering Experiment Station of the *University of Illinois,* under an agreement with the U. S. Bureau of Mines. The scope of the work, in part, was as follows:

1. Determination of the coefficient of friction for the flow of air in concrete ducts, as proposed for the tunnel.

2. Determination of the formulae for calculating the pressure necessary to deliver the air through concrete ducts of constant cross-section, with the air leaving or entering in uniform quantities at uniform intervals.

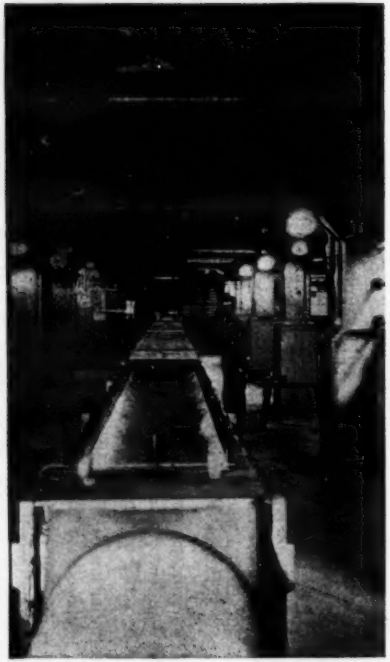


FIG. 7. GENERAL VIEW OF SMALL DUCT TESTED

FIG. 6 (Right). GENERAL VIEW OF LARGE CONCRETE DUCT TESTED

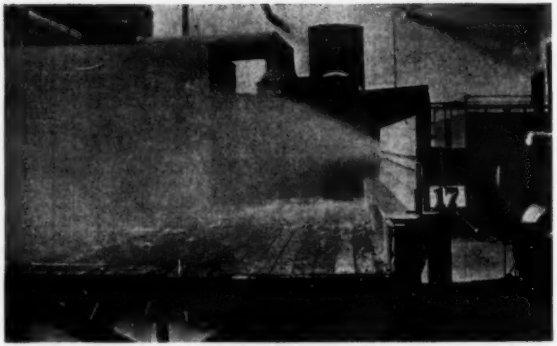


FIG. 8. FRESH AIR PORTS WITH EXPANSION CHAMBER

3. Determination of pressures necessary to convey air through various types of concrete elbows, louvres and in fresh air and exhaust air ports.

The experiments under items (1) and (2) were performed in a concrete model, the linear dimensions of which were one-half of those of the proposed duct under the roadway, and on a one-tenth scale metal and wood duct model. Fig. 6 is a general view of the large concrete duct. It was 300 ft long, with a wood duct 75 ft long connecting it to the test fan which had a capacity of 105,000 cfm and was driven by a 300 hp motor.

Outlets to half-scale dimensions of those proposed for the tunnel were provided on each side of the duct at uniform intervals throughout the 300 ft length. These outlets were provided with adjustable shutters for controlling the flow of air. The interior of the duct was smooth and free from fins. Three air measurement stations were provided, and the end of the duct farthest from the fan was equipped with adjustable gates for the control of pressures. Fig. 7 is a general view of small duct which was one-fifth the dimension of the concrete duct; its length was 60 ft and it was connected by means of a metal duct 29 ft long to a fan direct connected to a 10 hp d-c motor. The small duct was equipped with outlets similar to those of the large duct, and it also had three air measurement stations. Tests were run in the large duct to determine the coefficients of friction with various velocities of air flow, the inside of outlet ports being closed off flush with the inner duct surface.

A large series of experiments were conducted to determine the pressure required to expel varying quantities of air from the outlets for various duct lengths. In these tests the outlets had to be adjusted so that uniform quantities of air were expelled from each outlet. As a result of this study, a formula was developed for calculating the pressure required for supplying air to a duct from which the air is removed in equal quantities at uniform intervals along its length. After these tests were completed the fan inlets and outlets of the supply fan were reversed so that the air was drawn through the duct after entering the side ports, thus representing the exhaust air duct of the tunnel. These tests showed that much greater power was required to draw the air through the duct (corresponding to exhaust duct) than to force it through (corresponding to fresh air duct).

To minimize the power requirements for exhausting the air, studies were made to design a port which would be practicable to install in the ceiling slab of the tunnel and allow the air to enter the duct as nearly parallel to the main flow as possible, instead of at right angles. The small model duct was designed with a removable top. One top was solid, and with it on the duct all tests made on the large duct were duplicated. From these tests and a knowledge of the law of dimensional homogeneity, a formula was developed for the calculation of the pressure requirements to force the air through the duct.

After completion of the blower duct tests the solid top of the small duct was removed and replaced with another top in which various types of exhaust ports were investigated. From these experiments it was found that the most satisfactory port was one with an inclination of 45 deg in the direction of the main flow in the duct. With this type of port a series of tests was made from which a formula for the exhaust duct pressures was developed. From these tests the following formulae were developed:

Coefficient of friction:

$$f = a + \frac{b}{m^2 w^2} \tag{1}$$

where

$a =$ 0.0035
$b =$ 0.01433
$m =$ area of duct in square feet divided by perimeter in feet
$w =$ velocity of air in feet per second $= \frac{Q}{A}$
$Q =$ cubic feet of air per minute passing the end of the duct nearest the fan
$A =$ area of duct in square feet

For blowing condition:

$$\text{T.P.} = \frac{12y}{D}\left\{\frac{W_1^2}{2g}\left[\frac{aLZ^2}{3m} - \frac{1}{2}(1-K)Z^2\right] + \frac{bLZ}{2gm^5}\right\} + P_o \tag{2}$$

where

T.P. = Total pressure in inches of water
$y =$ Density of air in duct in pounds per cubic foot
$D =$ Density of water in pounds per cubic foot
$W_1 =$ Velocity of air entering the duct in feet per second
$g =$ 32.2 feet per second, per second
$a =$ 0.0035
$b =$ 0.01433
$L =$ Total length of duct in feet
$Z = \frac{L - X}{L}$
$X =$ distance from entrance of duct to any chosen section
$m =$ Hydraulic mean radius in feet = area of duct divided by perimeter
$K =$ 0.615, a factor allowing for turbulence in the duct
$p_o =$ Bulkhead pressure in inches of water

For exhaust condition:

$$\text{S.P.} = \frac{12y}{D}\left\{\frac{W_1^2}{2g}\left[\frac{aL}{(3+c)m}Z^3 + \frac{3}{2+c}Z^3\right] + \frac{bL}{2gm^5(1+c)}Z\right\} + p_o \tag{3}$$

where

S.P. = the suction head or negative static pressure
$c =$ 0.25 for flow in excess of 200 cu ft per minute per foot of tunnel. For lower rates, $c =$ 0.20

All other terms have the same significance as in equation (2).

The exhaust equation holds for all lengths of ducts, but that for blowing requires certain other considerations for duct lengths wherein the minimum static pressure falls outside of the main duct from which air is being taken off. The minimum static pressure in a duct must be sufficient to handle the maximum quantity of air to pass the port. To determine what these minimum pressures were for various quantities of air, tests were made on full size fresh air ports with expansion chamber, as shown in Fig. 8, and also on full size exhaust air ports, as shown in Fig. 9. The results of these tests were plotted and used in the setting of the ports in the tunnel.

In addition to the foregoing tests, other tests were made for the determination of pressure required for the passage of air through elbows of various sizes and shapes, and through louvres. After having calculated the pressure required to handle the air through an entire duct section, the air horsepower was calculated by using the formula:

$$\text{hp} = \frac{P \times Q \times 62.4}{33000 \times 12} \qquad (4)$$

where

hp = Horsepower
P = Pressure in inches of water
Q = Total quantity of air handled in cu ft per minute

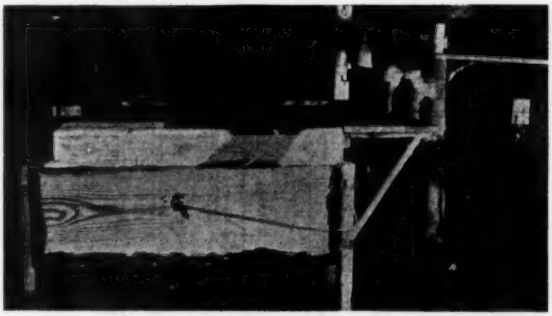


FIG. 9. FULL SIZE EXHAUST AIR PORTS

The total horsepower to be delivered by the motor was determined by applying the efficiencies of the fan, transmission and motor.

DEMONSTRATION OF THE OPERATION OF THE SYSTEM

After the completion of the investigations already described, it was necessary to demonstrate the operation of the proposed system on a large scale and also to demonstrate whether the air supply to the driveway should be from the duct over the driveway or the duct under it. This demonstration was conducted in the U. S. Bureau of Mines experimental mine at Bruceton, Pa., part of which was reconstructed for this purpose, providing a miniature tunnel, oval in plan, with a driveway length of 400 ft, located about 1,200 ft in from the mountainside and entirely shut off from the outside atmosphere except through a drift connecting the tunnel to the ventilation plant located outside. Fig. 10 shows a plan of the layout. The tunnel was 9 ft wide and 8 ft high, with a roadway sufficient for one line of automobiles. Ventilation ducts were constructed above the ceiling and under the roadway. Provision was made so that the flow of air to and from the tunnel could be reversed, using alternately the upper and lower ducts for the fresh air supply.

Tests were made in the experimental tunnel with as many as eight automobiles at a time, operating as they would in the proposed tunnel. Some tests

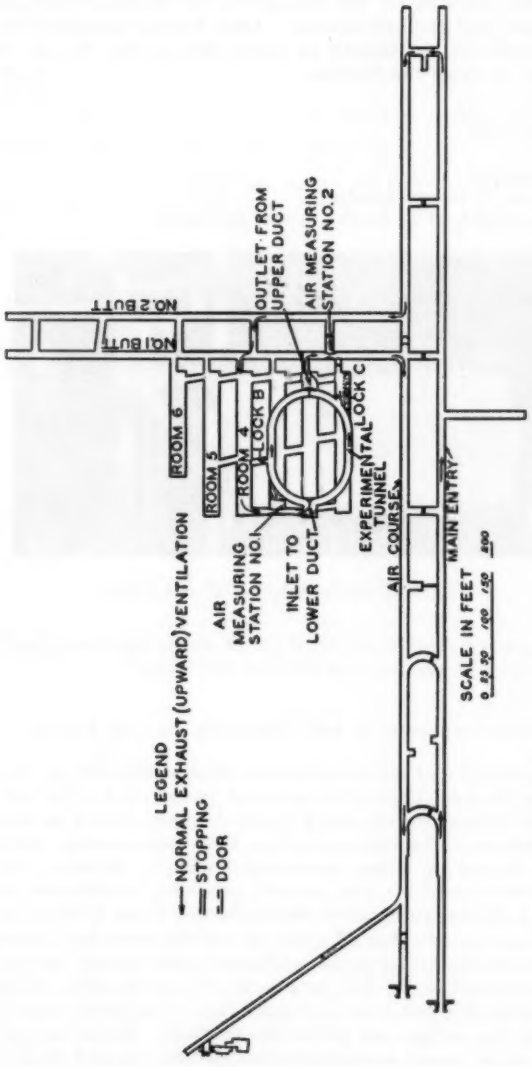


FIG. 10. LAYOUT OF U. S. BUREAU OF MINES EXPERIMENT MINE

were made with the fresh air supply from the duct under the roadway into the driveway just above the roadway, and exhausted through ports in the ceiling to the duct above. Others were made with the air supply from the duct above the ceiling into the driveway and exhausted through ports just above the roadway into the lower duct. Samples of air were taken from various points in the driveway and also from the moving automobiles, and analyzed.

The results of the tests showed that the two methods of transverse air movement were practicable for tunnel ventilation, and that the best method, from the standpoint of economy and safety against fire hazard, was the one in which the fresh air is introduced from the duct under the roadway and exhausted through the duct above the ceiling. With this method the CO passes the breathing zone of drivers once, whereas, with the reverse method, the CO passes twice, once on its way up from the exhaust pipe, and again on its way down due to the action of downward air currents from the upper duct.

The amount of CO given off by the automobile during these tests checked very closely the results previously obtained from the investigation of the amount and composition of exhaust gases from automobiles. The physiological effects of CO were again carefully checked during these tests; the results agreed closely with the findings of the earlier investigations.

Tests were also made to determine the effect of hot weather on the absorption of CO by humans. For this purpose the tunnel air was heated by gas jets to a dry-bulb temperature of 103 F and a wet-bulb temperature of 95 F. It was found that heat up to this degree had no effect on the rate of absorption of CO. Observations were also made on the flow of heat through the concrete walls of the experimental tunnel under conditions approximating those proposed for The Holland Tunnel.

The original investigation to determine the physiological effects of CO on passengers using the tunnel did not include a determination of the effect of long exposures to CO in concentrations up to maximum allowance, such as would be experienced by the employes on duty in the tunnel. To obtain this information, additional research work was done under an agreement with the Bureau of Mines at the Bureau's Experimental Station at Pittsburgh, Pa. The information obtained was used in the setting up of working schedules for the tunnel employes. The investigation was carried out in a gas-tight chamber especially equipped for the purpose.

Six subjects were simultaneously exposed in each test. Of these, three remained at rest and three exercised mildly. The exercise consisted of stepping up on a bench 24 in. high a number of times in rapid succession each half hour. The subjects were exposed to concentrations of CO ranging from pure air to 4 parts in 10,000 parts of air, for periods of from four to seven hours each day for 68 days in succession.

As in the previous investigations, the frontal headache was the first symptom and generally occurred when about 20 per cent CO hemoglobin saturation is attained. Fig. 5 shows the relation of the number and time of occurrence of frontal headache to percentage of CO saturation of blood, and also the rate of elimination of CO. These curves make possible the determination of the proper length of shifts for employees working in the tunnel under various percentages of CO in the air.

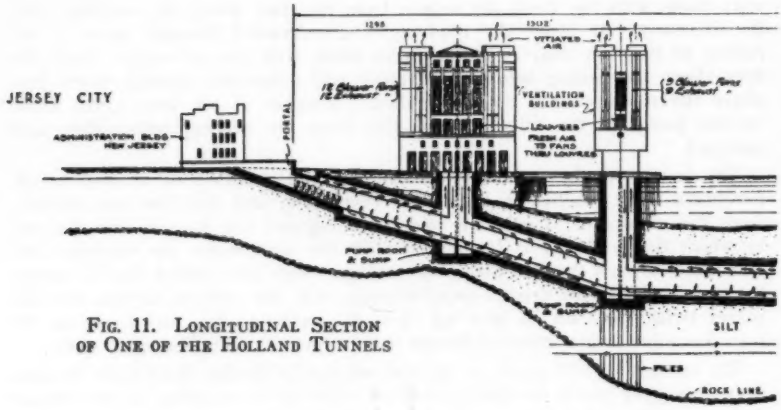
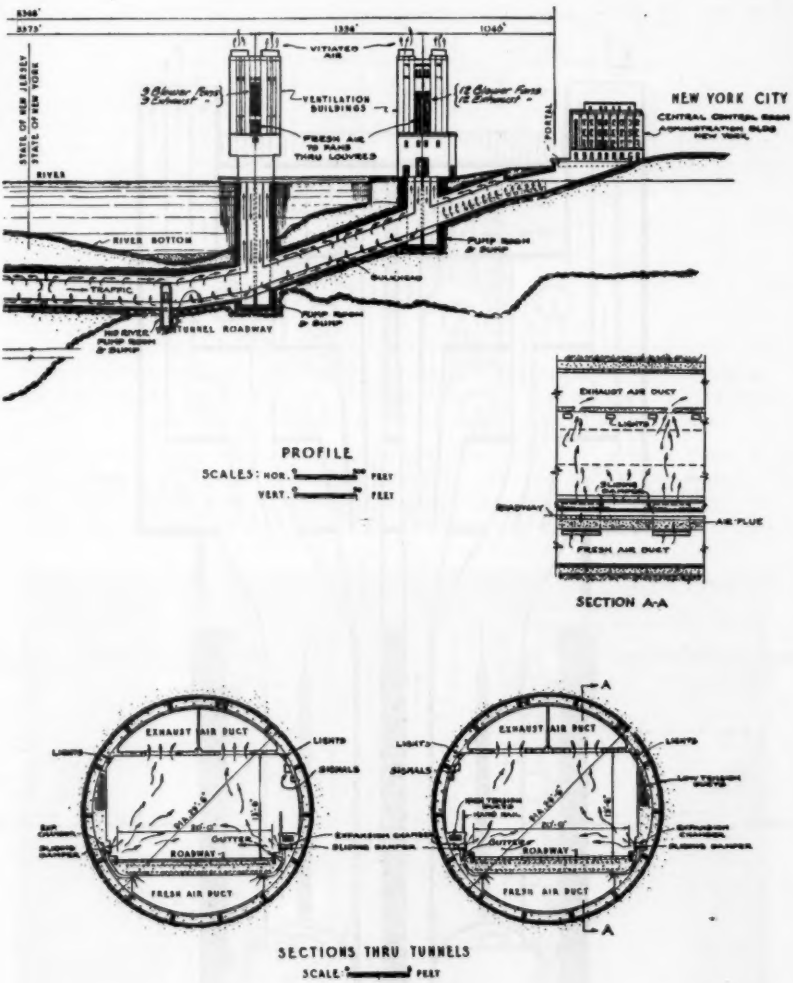


FIG. 11. LONGITUDINAL SECTION OF ONE OF THE HOLLAND TUNNELS

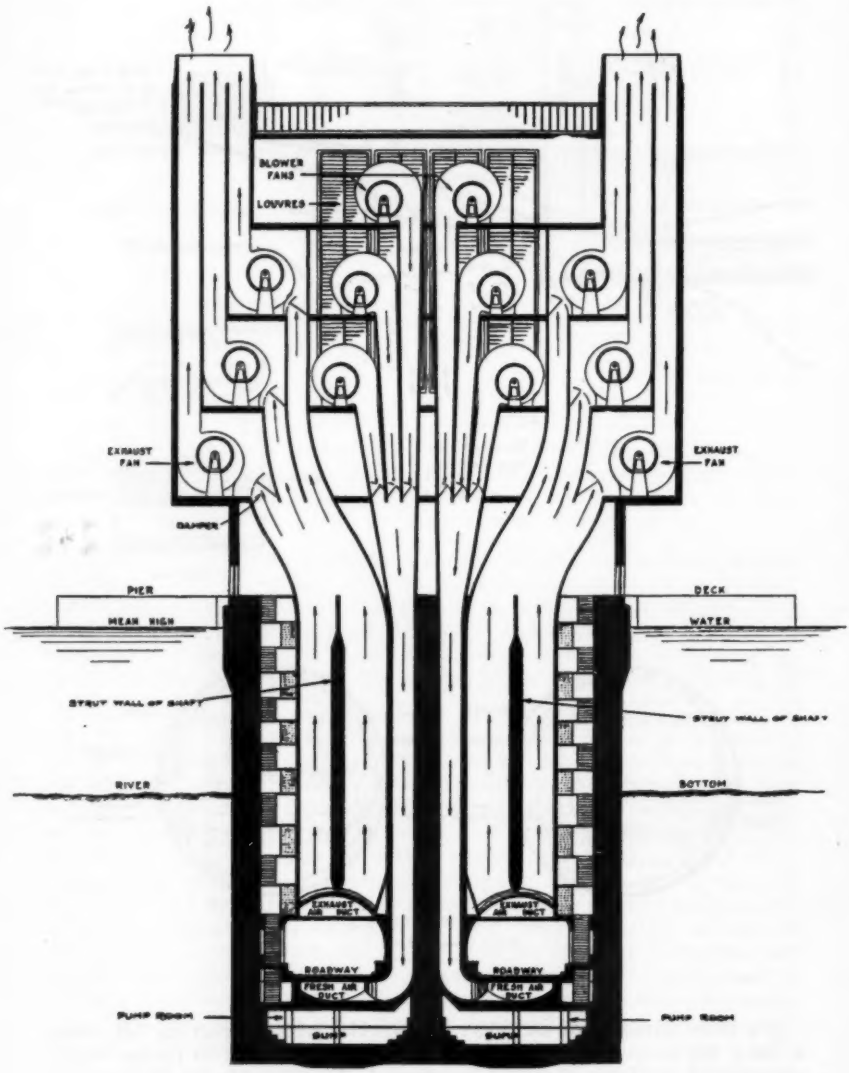
The investigation also showed definitely that the administration of oxygen after exposures to CO greatly diminished the time necessary for desaturation. In the operation of the tunnel the administration of oxygen has not been necessary as no symptoms of CO poisoning have developed.

The investigations show what can be done by the close co-operation between the engineer and the research scientist. The engineer analyzed the problem with which he was confronted, reduced it to its several component parts, and stated the different points on which research work was required. The scientist conducted the research work in such a way as to furnish the information on the various phases necessary for the solution of the problem as a whole.

The system of ventilation adopted for The Holland Tunnel is shown in Figs. 11 and 12. Fig. 11 shows a longitudinal section of one of the tunnels, with the four ventilation buildings. The fresh air duct is below the roadway, and from this duct air is taken off through flues located 10 to 15 ft apart in either side of duct. This air is delivered to the continuous expansion chamber on either side of the driveway. The face plate on these expansion chambers is adjustable and is set so that, when the maximum supply of air is delivered, the velocity of air through the slot is 1,000 ft per minute. In order to control the supply of air from the duct, slide dampers are provided in the expansion chambers on top of the flues. The vitiated or exhaust air duct is above the ceiling slab, and the air from the driveway enters this duct through exhaust ports located 10 to 15 ft apart. These exhaust ports are special in design, being made of cast iron ⅜ in. thick and provided with an adjustable zinc slide. Each tunnel has seven fresh air ducts and seven exhaust ducts, and each duct terminates in three openings, one to each fan. Thus, there are 21 fresh air fans and 21 exhaust air fans to each tunnel, or a total of 84 fans for the two tunnels. These fans are located as follows: 12 exhaust fans and 12 fresh air fans in each of the land ventilation buildings, and 9 exhaust fans and 9 fresh air fans in each river ventilation buiding.



The three terminals of each duct are provided with dampers so that, when a fan is not in operation, the duct terminal may be shut off to prevent short-circuiting of air from other fans, as shown in Fig. 12. Air for the fresh air fans is drawn in through the louvres in the sides of the buildings, and the vitiated air from the tunnel is exhausted through the stacks at the corners of the buildings.



SECTION
TYPICAL RIVER VENTILATION BLDG

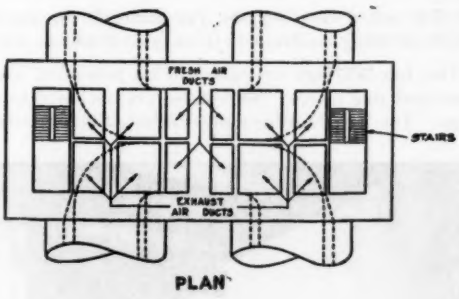


FIG. 12. SECTION AND PLAN SHOWING METHOD OF VENTILATION OF THE HOLLAND TUNNEL

The ventilating fans are backward-curved blade type of fan, with a high static efficiency and a non-overloading horsepower characteristic. The housings are double width and double inlet, constructed of No. 10 gage steel, and are split so as to permit of ready dismantling for repairs.

Fan bearings are self-aligning type, and are made with a high grade of Babbit metal. The thrust collars on fan shafts are machined on the shaft and bear against Babbitted surfaces in the bearings. Inspection doors are provided and, in the case of exhaust fans, the housings are connected to drains

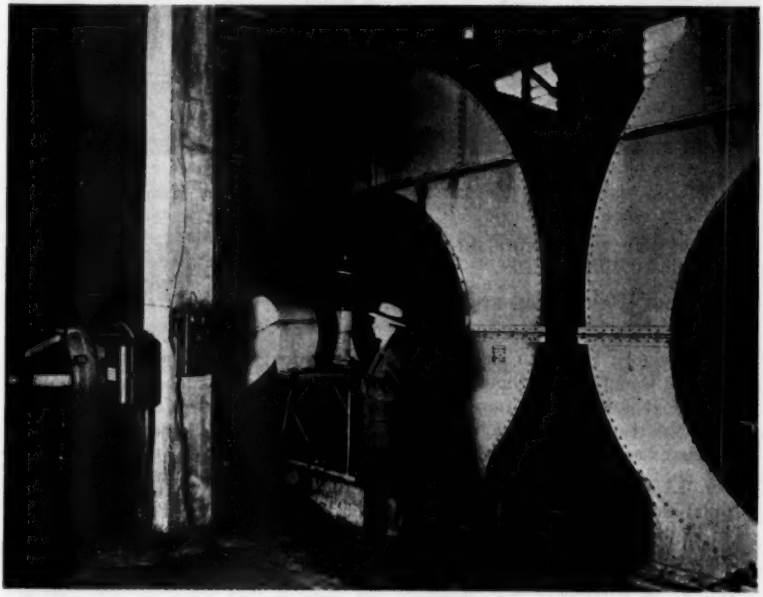


FIG. 13. TYPICAL OUTSIDE AIR FAN

so that water entering the fan through the exhaust stacks may be drained off, there being no dampers in exhaust stacks to prevent entry of water or snow.

The fan bearings are mounted on pedestals, which in turn are mounted on machined sole plates. Sole plates are anchored to steel structures of the buildings. The fan housings are anchored to the concrete floor. The fans are

FIG. 14. EXHAUST FAN INSTALLATION

driven by slip ring motors through chain transmission. The chains run in fully enclosed oiltight cases.

Motors of 100 hp or less are 440-volt, 60-cycle, 3-phase; others are 2,200-volt, 60-cycle, 3-phase.

Each exhaust fan is in an airtight chamber along with its transmission, but the motor is outside of the room. These exhaust fan chambers are provided with removable steel partitions which were erected after the fan was installed, and can be readily removed in case of fan renewals. Exit and entrance from these chambers is by air locks.

Fig. 13 shows a typical fresh air fan, and Fig. 14 shows an exhaust fan installation. The fan motors can be operated either from the main control room in the New York Administration Building, from the local control board in the ventilation buildings, or from the local controls near each motor.

When the equipment was installed, the control was designed for two accelerating speeds and three running speeds. The three running speeds were

approximately ½, ¾ and full speed. The control system is now being changed to include two more running speeds between ¾ and full speeds, to effect further economy in the operation of the system.

CHARACTERISTICS OF THE VENTILATION SYSTEM

The characteristics of the ventilation system are as follows:

1. Uniform distribution of fresh air throughout the length of the tunnel.
2. The ventilation of the tunnel is not affected by outside atmospheric conditions.
3. Air supplied to or exhausted from the driveway can be definitely controlled.
4. The exhaust gases are quickly diluted and removed.
5. There are no high velocity air currents in the driveway due to the air supply.
6. Smoke from fires in the tunnel is quickly removed and prevented from spreading.

To operate the 84 fans require 6,120 hp.

To handle the maximum quantity of air for normal ventilation requires 56 fans, which require 4,100 hp; 28 of these fans are fresh air fans which supply 3,761,000 cfm.

DISCUSSION

A. C. FIELDNER (WRITTEN)[1]: The author has stated that the solution of the ventilating problems connected with the Holland Tunnels showed what could be done by close cooperation between the designing engineers and research scientists. As representing the latter group who were fortunate enough to be connected with this pioneering project, I wish to corroborate this statement.

A problem clearly stated is half solved. The late Mr. Holland and his associates came to the Bureau of Mines with clear-cut questions which could at once be attacked by the experimental method and definite answers obtained.

I have never been connected with any research problem which was undertaken and conducted with keener interest or greater enthusiasm. Everyone appreciated the fact that he was engaged on fundamental pioneering research, the results of which would become classic in the annals of vehicular tunnel ventilation, and would be applicable not only to the Holland Tunnel in New York but many other tunnels that would be constructed in other places and under other conditions. Great credit must be given to Mr. Holland for initiating tunnel ventilation research and to the Tunnel Commissions of New York and New Jersey for supporting this research on a broad basis and making it public for the benefit of other projects to follow. The fundamental way in which these investigations were carried through has made the results of great value in many related fields, particularly the work on carbon monoxide.

At the time the investigation was started, methods for determining carbon monoxide in air or in the blood were slow and tedious. It was necessary to develop rapid and accurate methods for determining carbon monoxide in air contaminated with exhaust gases, to review and improve upon the methods for determining carbon

[1] Published by permission of the Director, U. S. Bureau of Mines. (Not subject to copyright.)

monoxide absorbed in the blood, and to develop an automatic indicating and recording device to aid in controlling the ventilation of the finished vehicular tunnels. All of these objectives were attained. The pyrotannic acid method [2] was developed into a quick and accurate procedure for determining carbon monoxide in blood. A simple, compact field apparatus, which can be carried in the coat pocket, was devised. The method requires only a drop of blood and can be used also for determining small percentages of carbon monoxide in the atmosphere.

It was early realized that an indicating and recording device was very desirable and practically essential for controlling the ventilation in the completed vehicular tunnel. Work was begun at the Bureau of Mines in devising such an apparatus. The most promising method appeared to be the utilization of the heat of combustion of the carbon monoxide. This method in principle had already been developed during the war in analyzing carbon monoxide-air mixtures used in developing catalysts for the carbon monoxide gas mask.[3] It was thought that this particular principle, of measuring with a thermocouple the rise in temperature produced by the oxidation of carbon monoxide, could be made applicable to very small percentages of carbon monoxide as well as the comparatively large ones which had been used in testing the gas masks. In order to secure the required degree of sensitivity, it was found necessary to use multiple thermocouples, the final design having 48 copper-constantan couples in series. These were kept at a uniform temperature by being surrounded with a bath of boiling water. One junction of the thermocouples was placed at the entrance, and the other junction at the exit of the catalyst chamber. With 48 thermocouples in series the apparatus was found sensitive enough to positively indicate 1 or 2 ppm of carbon monoxide in air.[4]

This recording apparatus has been found useful not only in controlling the ventilation of tunnels, but also in testing gas stoves for carbon monoxide. In the Testing Laboratory of the *American Gas Association* at Cleveland, O., and in the testing laboratories of a number of the large gas companies this apparatus is regularly used for measuring the amount of carbon monoxide, if any, given off from room heaters.

In securing the answer to problem 1, the amount of carbon monoxide given off by various automobiles and trucks, reliable information was obtained for the first time on the actual efficiency of combustion in automobiles and trucks as operating on the streets. It was found that adjustments of carburetors varied over the entire scale from the leanest possible mixture at which a car would operate to the richest mixture, and in fact the average percentage of carbon monoxide for all automobiles and trucks tested was very close to the mean of the maximum possible variation at which an engine would run. It is probable that knowledge of the energy lost through carbon monoxide in exhaust gases has stimulated inventors to improve carburetion and combustion characteristics in automobile engines, although no similar tests have been made to show whether the average efficiency in actual operation has increased. With the present low prices of gasoline, it is unlikely that much attention has been given to increasing thermal efficiency. The public is more interested in flexibility and smoothness of operation which are obtained with mixtures on the rich side. There has, however, been some interest in the development of carbon monoxide eliminators which automatically burn the carbon monoxide in the exhaust gases before they are discharged into the atmosphere. Such positively acting and foolproof eliminators, if developed, would be useful indeed in reducing the amount of ventilation required in vehicular tunnels and in reducing the amount of accidental asphyxiation from automobile exhaust gases.

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[2] Sayers, R. R., and Yant, W. P., The Pyrotannic Acid Method for the Quantitative Determination of Carbon Monoxide in Blood and in Air, *U. S. Bureau of Mines Technical Paper 373*, 1925.

[3] Lamb, A. B., and Larsen, A. T., The Rapid Determination of Carbon Monoxide in Air, *Journal American Chemical Society*, Vol. 41, December, 1919, pp. 1908-1921.

[4] Fieldner, A. C., Katz, S. H., and Meiter, E. G., Continuous CO Recorder in the Liberty Tunnels, *Engineering News-Record*, Sept. 25, 1925, pp. 423-4, *U. S. Bureau of Mines Technical Paper 355*, 1926.

Aside from these important developments in the technique of chemical and physiological research involving carbon monoxide, basic information, not previously available, was obtained on the physiological effect of exposure under various conditions to small percentages of carbon monoxide. Such information is of wide application because of the almost universal danger of exposure to this ubiquitous gas in our modern environment. In the home we may be exposed to carbon monoxide from leaking manufactured gas or products of incomplete combustion; in the garage to automobile exhaust gases; in industry to water gas, producer gas, blast-furnace gas, and incomplete combustion products.

These are some of the more important applications in various fields of some of the results of the cooperative tunnel ventilation investigations. Such wide application of the data obtained would not have been possible if the designing engineers of the Holland Tunnel had been satisfied with a few empirical tests followed by some engineering guesses fortified by a large factor of safety.

YANDELL HENDERSON (WRITTEN): The Holland Tunnel is among the supreme achievements of modern engineering. Perhaps each separate element in the problem was not greater than other large engineering work has involved. But the planning of the diverse elements and the skill and thoroughness shown in bringing them into their proper relations one to another, so as to produce a unified whole, was a supreme achievement. Above all others the credit for this achievement belongs to the late Clifford M. Holland, the chief engineer. It is wholly deserving that the tunnel should bear his name.

It has often been said that for the Panama Canal the solution of the problems of malaria and yellow fever, and the control of mosquitoes were as essential as the surveys or the designing of the locks. The digging also depended on the management of the trains of cars which conveyed the dirt from the excavation to the dumps, and these were essentially railroad problems. Thus there were many elements other than the primary problem in the construction of the Panama Canal. Similarly in the construction of the Holland Tunnel there were physiological problems, chemical problems, problems of automotive engineering, and problems of ventilation in addition to the problem of driving the largest submarine tunnel that had ever been attempted. Clifford M. Holland saw his problem whole with every part in its place. He assigned each particular problem to an expert in that particular field. He and his able associates then combined all the factors. As a result they produced a work perfect in every detail.

The fundamental investigations for the Holland Tunnel will never have to be done over again. The data collected apply equally well to all other similar undertakings everywhere. When a tunnel was planned two or three years ago, under the Mersey River at Liverpool, England, a committee of the House of Commons held a hearing at which Dr. J. S. Haldane, the great English authority on carbon monoxide and other gases, merely presented the data that were collected for the Holland Tunnel. He assured the committee that these data supply all that is essential to assure perfect safety to the British public. The Mersey tunnel is now being constructed on the basis of those data, and will be ventilated according to their requirements.

When the city of Wellington in New Zealand, recently planned a tunnel for motor vehicles under a river there, the engineer wrote to me and I sent him the data that we had collected for the Holland Tunnel. That tunnel in New Zealand will be constructed and ventilated on the basis of the data which we collected for the Holland Tunnel.

In all aspects the thoroughness of the preparation for the Holland Tunnel stands as one of America's outstanding achievements in every department of engineering, sanitation and chemistry that was in any way concerned. This tunnel demonstrates how the solution of a great engineering problem and how the application of scientific data to a great public work may be done supremely well.

S. H. KATZ [5] (WRITTEN): Mr. Davis' review of the Development of the Ventilation System of the Holland Tunnel brings out in the clear light of retrospect the numerous problems requiring solution and the specific data nonexistent when the undertaking began. The solutions bulked large in the design of the tunnel, now a model of its kind. Time and experience have proved the ventilating system fully adequate in providing comfort and affording health and safety to the public now using the Tunnel.

It was my privilege to study the atmospheric conditions in the Holland Tunnel under direction of the Bureau of Mines and the Superintendent of the Holland Tunnel during its early operation.[6] The first day of public use, Nov. 13, 1927, provided an excellent test not only of the physical operation but also of the reliability of the data and the engineering forethought described by Mr. Davis that went into the physical equipment. On that day 51,750 automobiles passed through the tunnel, a figure not exceeded until recent expansion in traffic between New York and Jersey City induced partly by the tunnel itself. Public interest in the tunnel was the principal reason for the big traffic of the first day. A maximum of 2,206 automobiles per hour (2-3 P.M.) passed through the North Tunnel and 2,371 through the South Tunnel (4-5 P.M.). This approaches closely the maximum of 2,540 per hour estimated maximum capacity.

The ventilating air supplied during the first day was ⅔ of the maximum; the highest concentration of carbon monoxide determined at any time by continuous carbon monoxide recorders [7] operating one on each of the 14 separately ventilated sections of the tunnel was 3.2 parts per 10,000. This was comfortably under the 4.0 parts per 10,000 maximum tolerable concentration of carbon monoxide.

Comparison of the quantities of the carbon monoxide evolved in the tunnel during the first two days with those calculated from data of the early tests [8] showed a maximum difference of 29 per cent more carbon monoxide in a single ventilation section of the tunnel compared to that calculated. This occurred on the second day of operation. The difference may be due partly to the greater grade in the tunnel 3.48 per cent as compared with that used for the early experiments, 3.00 per cent, and also to the increased speeds in the tunnel, maximum 30 mph compared to 15 mph in the early experiments.

The agreement is fully as good as may be expected in a large engineering undertaking encountering such variable conditions as the run of automobile traffic. That the problems, as related by the author, were fully foreseen, well analyzed and correctly solved is proved by the tunnel in operation.

Mr. Davis stressed the importance of the cooperation of research scientists in the solution of problems analyzed and specified by engineers. The relationship was a happy one in developing the ventilating system of the Holland Tunnel. Similar relations will bear prolifically in numerous other engineering undertakings needing basic data for purposes of planning and design.

G. A. POSEY (WRITTEN): The paper by Mr. Davis is interesting in setting forth the actual methods employed in developing the so-called *transverse method of ventilation*. Having no precedent to guide them, Mr. Holland, the chief engineer, and his staff performed real pioneering work and they are to be congratulated on evolving such an adequate method.

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[5] Assistant Chief, Protective Development Division, Chemical Warfare Service.

[6] Katz, S. H., and Frevert, H. W., Chemical Control of Ventilation at the Holland Tunnel, *Industrial & Engineering Chemistry*, Vol. 20, 1928, 564-70.

[7] Katz, S. H., Reynolds, D. A., Frevert, H. W., and Bloomfield, J. J. A Carbon Monoxide Recorder and Alarm, *U. S. Bureau of Mines, Technical Paper 355*, 1926, 35 pp.

[8] Fieldner, A. C., Straub, A. A., and Jones, G. W., Amount and Composition of Automobile Exhaust Gas, *Journal American Society of Heating and Ventilating Engineers*, Vol. 32, 1926, pp. 17-36.

After a thorough investigation I adopted the same system of ventilation for the George A. Posey Tube. The method both in the Holland Tunnel and the George A. Posey Tube has proved to be a success. Especially was it necessary here to employ a method of ventilation that would be effective in a single-bore tube, carrying single lines of traffic in different directions. Every moving vehicle in a tube sets up a current of air because of its own movement and vehicles in opposite directions bring into being currents of air in opposite directions, so for single-bore tunnels, to obviate the effect of fires and smoke, *the transverse method is the only one that is really efficient.*

I have found, however, due to better construction of vehicles, probably that the amount of carbon monoxide produced by all kinds of automotive equipment now is less than the original experiments show for similar grades of roadway.

Considering the dilution of exhaust gas, necessary in a tunnel for human safety, two things must be considered: *First,* the dilution necessary for the travelling public, who are supposed to remain in the tunnel only a few minutes; *Second,* the police, maintenance men, or operators who must remain inside the tunnel for hours at a time.

The experiments indicated that 4 parts of carbon monoxide in 10,000 parts of fresh air could be endured by human beings for a period of 1 hr. Consequently this ratio was made the basis of calculation for quantities of necessary fresh air for both the Holland Tunnel and the George A. Posey Tube. I have found here that where the air quanities were such as to give that ratio of carbon monoxide that the smoke or unburned carbon was so great as to be objectionable to the occupants of vehicles and injurious to the traffic officers. It was necessary to reduce the ratio to 1 part in 10,000 and that has been our standard for the past 1½ years. Even with this amount of carbon monoxide actual tests show that the traffic officers who work in 8-hr shifts are showing symptoms of carbon monoxide poisoning, so much so that I am now constructing booths inside the tube, directly connected to the fresh air duct, where traffic officers on duty will be stationed most of the time. These booths will be ventilated, lighted and heated.

It would be in order now, however, to obtain the actual quantities found to exist under actual running conditions and I am hoping that Mr. Davis will supplement his fine paper by giving data on the effect of small amounts of carbon monoxide on human beings over a long period of time.

Efficient mechanical means of ventilation has made vehicular tubes and tunnels workable in this day of the automobile and until some means of eliminating the carbon monoxide from the exhaust gases these mechanical means will be a necessity.

There can be no doubt but that the dilution of carbon monoxide as the basis of dilution of fresh air in vehicular tubes is correct, but the amount of carbon monoxide must be decreased in order that the other vapors that will produce irritation to the eyes, nose and throat of either the users or traffic men in a tube or tunnel must be taken into account.

G. S. RICE[9] (WRITTEN): This paper is so complete in describing the successive steps in the development of the unique method of ventilation of the Holland Tunnel that it leaves little to be said. I will, however, take advantage of the occasion to make some remarks from the point of view of mining in which ventilation is a fundamental factor whether it be in coal mining or in deep hot metal mines or in tunneling through rock for various purposes. Inflammable gases are not confined to coal mines. Several disastrous explosions have occurred in California during the last year in driving tunnels through hard rock formation. There is opportunity for the application of some of the principles of ventilation determined for the Holland Tunnel.

[9] Chief Mining Engineer, U. S. Bureau of Mines, Washington, D. C., and former Chairman of the Ventilation Committee of the *A. I. M. & M. E.*

It is not an exaggeration to say that the Holland Tunnel presents one of the most ideally carried through engineering undertakings of all time and is a project which will be regarded as an engineering classic. Its successful achievement reflects the remarkable engineering skill and administrative ability of the late Clifford M. Holland and members of his staff, especially Ole Singstad and A. C. Davis. Nothing was left to chance. Every detail of its complicated, difficult design of construction, machinery and operation was worked out in advance, and this is an undertaking which in certain fundamental matters had no precedent to give guidance.

This project involved three branches of engineering, civil, mining and mechanical, but chemical, physical and physiological questions had first to be determined. Finally and not least, the official agencies of the two states and an influential part of the public, had to be shown, in spite of strong opposition, that the project was workable and the great expense justified.

Few persons who pass through the tunnel, quickly and easily, while they admire the splendid construction and possibly note that the air is better than on congested city streets, have any realization that the ventilation method and control is fundamental to the successful operation of the tunnel, or that the methods were so novel and without precedent that it took several years of systematic research culminating in assemblage tests with automobiles in a tunnel of facsimile section, where chemical, physical and physiological studies were made.

The by-products of the various researches carried on for the Holland Tunnel, such as the physiological effects of carbon monoxide, including the time for recovery from inhaling small amounts, the chemical constituents of motor exhaust gases, the rate of interchange of heat through walls, the determination of the coefficients of friction of air currents at different velocities through different kinds of ducts, orifices and bends, and the rapidity of diffusion of gas under special arrangements, are all of great value not only in future vehicular-tunnel ventilation problems but likewise in mine ventilation and in long drifts driven above water level.

With pleasure and professional pride I recall my early association with Mr. Holland and members of his staff. In 1919, accompanied by Dr. Edward Levy, physiologist, with whom I had previous official contacts on the caisson disease question, these men came to my office at the Bureau of Mines in Washington, to request the assistance of the bureau for a research in physiological effects and the character and quantity of motor exhaust under various conditions. Dr. Van H. Manning, then director of the bureau, authorized the investigations and asked me to arrange for the cooperation.

Dr. Yandell Henderson of Yale University, who was then the consulting physiologist of the bureau, was asked to undertake the research on the effect of exhaust gases for different concentrations and times of exposure. He began these investigations on Nov. 20, 1919. A. C. Fieldner, then chief chemist and superintendent of the Bureau's Pittsburgh Experimental Station, was given charge of the determination of the quantity and character of exhaust gases from different types of motors under different conditions. This work began on Dec. 1, 1919, and continued for nearly a year.

When research on these two problems was first requested by Mr. Holland in 1919, he told me that he was not going to consider details of the ventilation method from an engineering viewpoint until these two problems had been reported upon. However, he discussed the general features of the requirements of construction and the several ventilation proposals that had been made. One of these employed a fresh air duct below the roadway under fan pressure and a space above the roadway for the collection of exhaust gases data, or the arrangement might be reversed. This method he contrasted with longitudinal ventilation urged by certain prominent engineers as being more practical and less expensive to operate.

At that time, when we visited experimental wind tunnels of the Bureau of Standards and U. S. Navy Yard to observe their test methods, in the course of the discus-

sion I urged that the first method be adopted, now known as the transverse method, supplemented by connecting the exhaust or "return," with suction fans in addition to pressure fans for the fresh intake air. This method would avoid the great hazard of spread of fire that occurs with the other method—longitudinal ventilation—which would menace persons in cars ahead, by smoke and gases carried forward by a strong air current. Also that there would be the further advantage in giving equally pure air in all parts of the tunnel, as described in the paper by Mr. Davis.

After the conclusion of the first two problems, two others were to be taken up by Mr. Holland with other organizations than the Bureau of Mines; however, I was able to present projects with him that led to his further cooperation with the bureau. Problem 3, Determination of air current movements and frictional constants, was taken up through the bureau with the University of Illinois Experiment Station, the problems being carried on under Prof. A. C. Willard.

Problem 4 was a multiple one and embraced a set of separate but related problems. I proposed and submitted plans for a test tunnel in the Experimental Mine near Pittsburgh, the general direction of the regular technical work of which came under my supervision. With the approval of the Director of the Bureau of Mines, other experimentation in the mine was set aside for about a year to undertake this most extensive work, which on the physiological side involved checking under practical conditions, the earlier work and also other phases. This part of the work came under the direction of Dr. R. R. Sayers, of the Public Health Service, who had been detailed to the bureau as its chief surgeon.

The other parts of the problem were conducted under the direction of Mr. Fieldner, who assigned different parts of the bureau. In the many tests there were over 50 persons engaged, chemists, physicists, physiologists, engineers, and assistants.

At all times Messrs. Holland, Freeman, Singstad, Davis, and Dr. Levy of the Holland Tunnel staff kept in close contact with the work by frequent visits.

The results of the researches were embodied in the Holland Tunnel, which, by its great success, reflects credit on all who had a part in the research work, as well as on its constructors. As concerns the general applicability of the ventilation system evolved for the Holland Tunnel, it has been adopted in full in two underwater tunnels since built, the Oakland-Alameda vehicular tunnel in California, and the Detroit-Windsor tunnel. There should, I believe, be an opportunity for using the transverse feature at least in perhaps a less elaborate form in the several highway tunnel projects now under construction or being planned. Also in certain respects the system in principle should be used in long mine drifts or tunnels above water level but which encounter gas in driving.

RALPH SMILLIE (WRITTEN): The products of combustion which constitute the major needs for ventilation of vehicular tunnels, namely, carbon monoxide and hydrocarbons, represent losses in the fuel consumed due to incomplete combustion. There is thus a two-fold economic loss, first in the production of toxic and irritating gases, as well as the cost of power consumed and the fixed charges and maintenance of equipment necessary to furnish air to dilute them. The most striking point in connection with the tests on the composition of exhaust gases was the wide variation between maximum and minimum carbon monoxide content, namely, from 0.5 to 14 per cent. These tests show that under average conditions, nearly 30 per cent of the total heat of the gasoline used is exhausted in the form of combustible gases, these gases being largely composed of asphyxial or irritating elements. The maximum thermal efficiency in an automobile engine is obtained when a slight excess of air is used over that required for complete combustion. For such a carburetor setting the horsepower is about 15 per cent less than the maximum, and operation is unsatisfactory until the engine is heated up, which is the reason for the average carburetor adjustment.

The author has clearly outlined the methods of determining the requirements of vehicular tunnel ventilation, and this paper should serve as a guide to those engaged in kindred problems. Although the tests on the composition of exhaust gases were made 10 years ago, before the general use of high compression engines and so called high test and tetra ethyl lead gasolines, the results appear to be close to present day conditions, and the enormous economic loss due to incomplete utilization of fuel in motor vehicles with its attendent liberation of poisonous gases will doubtless continue until the public is awakened to the benefits to be derived from more efficient carburetion.

PROF. A. C. WILLARD: I wish to speak on certain aspects of this paper which Mr. Davis has presented so briefly. The staff of the Tunnel Commission, headed up at the time this work was first conceived by Mr. Holland, the first chief engineer, and Ole Singstad, the present chief operating engineer, and Mr. Davis, the mechanical engineer, to my mind is an unusual group of men. We talk in this organization a great deal on scientific facts, not so often on personalities, and I wish to offer my testimony to the personalities and appreciation of those personalities who were charged with this undertaking.

Mr. Holland himself was one of the most far-visioned engineers I have ever met. He appreciated the importance of as much basic data for his engineering work as any engineer I have ever known and he was certainly admirably seconded by Mr. Singstad and Mr. Davis.

Our connection with the matter at the University of Illinois dealt with, as Mr. Davis indicated, the power requirements of the tunnel. Mr. Davis himself was the representative of Mr. Holland at the University throughout the period of 5 or 6 years when our part of the studies was being made. Never have I worked with a man who was so thoroughly appreciative of the problems that his chief had in his mind and went into such details as Mr. Davis did, to see that we obtained the data that Mr. Holland considered essential in the design and for the installation of the apparatus in this tunnel. Probably, there has been no phase of this investigation more important than the experimental work which was conducted with the two conditions of air supply and removal.

One problem was to find out definitely, how much power was required to get the air through the Tunnel ducts and into the roadway, and the other problem was to find out how much power it took to get the air out of the roadway and through the tunnel ducts again. It would seem that if you moved the same quantity of air in both cases you might have about the same power requirements.

In the experimental work at the University of Illinois it was discovered that the exhaust conditions were far more serious than the supply conditions. The power requirements to pick up the vitiated air through the ceiling exhaust ports was far greater than the power requirements to bring in air in the same quantity and blow it through the supply ducts. The reason for this is that, as the air entered each one of the slots in the ceiling that jet of air shot across the main stream of air and acted as an air screen, or as an air baffle to interfere with the current of air that had been picked up from the ceiling slots which were nearer the bulkhead than the slot in question. These velocities increase rapidly on approaching the fan and these screens became more and more of an obstruction, with the result that in the early experimental model Mr. Davis discovered that it required 2½ times as much horsepower to exhaust the air from the experimental model, with the first crude slots that were used, than was necessary to supply the same quantity of air through the same experimental length of duct. It was only through further experimentation, not anticipated in the original testing program, on the character of these slots that we found it possible to so design them that the power requirement on the exhaust side was reduced to practically the same as the power requirement on the inlet side.

That one phase of the experimental work more than justified, I believe, the entire cost of the experimental work on the tunnel ventilation. In other words, the project might have been a complete failure if we had not known this condition existed, and if the outlet slots had been designed in a very simple, conventional fashion similar to the inlet slots of the tunnel.

Through the proper design of the exhaust ports we were able to reduce the turbulence that existed in the exhaust duct to a point where the power requirements are approximately a balance at the present time between the supply and exhaust side. This difficulty developed because of the unusual length of the exhaust ducts, and the fact that there were many openings in each one of these long, horizontal exhaust ducts which made it possible for these incoming jets of air to seriously interfere with the air flow in the straight duct.

I know of no experimental project with which I have been connected where the engineers responsible for the design were so willing and eager to make use of every bit of experimental evidence that would improve in any way the operation of the tunnel. The 29-ft diameter of this tunnel was increased in one case 6 in. and in another case 1½ ft as the result of the ventilation studies on the power end alone. It was shown that the saving in operating costs would justify that increase in the tunnel size.

The matter of credit in a project of this character is so broad, it includes so many people, that it is difficult indeed to place credit anywhere except to say that everybody did his part.

I wish to call attention to one thing, with Mr. Davis' permission, and that is that the frictional equations for the supply ducts and the exhaust ducts, which are in the paper, are the result of a mathematical analysis made by Prof. G. A. Goodenough of the University of Illinois, now deceased, who took the experimental data for all sizes of models, for all lengths of ducts, for all conditions of air flow and reduced those experimental data to two fundamental flow equations which are in the paper to cover projects of a similar character. We already have and will have in the future many similar projects where we are ventilating long tunnels, subways or passages with many points of inlet and outlet for the air. I considered these two equations to be a most valuable contribution of a high scientific order to the art of ventilating or air handling and the credit for these two equations belongs distinctly to Professor Goodenough.

DR. R. R. SAYERS: I think that Professor Willard has told you a great deal and his statement about Mr. Holland gives an accurate idea of the broad vision of this unusual engineer. The tunnel problems were complex but he successfully outlined the research studies required and they were carried out in several places.

The investigations for the Holland Tunnel were made at the University of Illinois, Yale University and the Bureau of Mines and were vital to the final success of the engineering work that was divided by Messrs. Holland, Freeman, Singstad and Davis.

The last report of the studies made by the U. S. Bureau of Mines was made in March, 1929, on the length of time that men could be exposed to different concentrations of carbon monoxide. This did not apply to persons passing through the tunnel as that had been worked out long beforehand; but was intended for the people who were on guard.

Concentrations of 2 parts in 10,000, 3 parts in 10,000 and 4 parts in 10,000 were selected and we exposed a group of people from 2 to 6 hr a day to those concentrations for 68 days, 7 days in the week. It was found that 2 parts in 10,000, in 3½ to 4 hr will cause slight symptoms of frontal headache, and that 3 parts in 10,000 will cause similar reaction in 2½ hr; 4 parts will cause this reaction in 1½ hr and, if the exposure is a little longer, say 30 min to 1 hr, a slight occipital headache may result.

These are important figures and they are valuable to all of us. They apply not only in tunnel ventilation but in homes and public buildings.

Of value to ventilating engineers that was developed in this research is an instrument that will analyze and record the amount of carbon monoxide in the air at all times. It was a by-product, if you will, of the study. I believe there were 14 of these instruments installed in the tunnel and they keep a constant record of the amount of carbon monoxide present in all parts of the tunnel. There have been 139 fires in a year in the tunnels and it was easy to see on the record when every one occurred, because a little bump appeared on the sheet when there was an increase in carbon monoxide. I think we need to pay tribute to the men who organized this work. It is of great value to the public.

A. C. DAVIS: I could talk for several hours regarding the tunnel. The subject has many ramifications and I covered only a few of them in my original outline of the paper. The studies outlined by Dr. Sayers on the effects of *CO* on our employees have indicated that we obtain the most economical results and the least number of complaints from our patrolmen when they remain in the tunnel 2 hr and then come out for 2 hr, then repeating the same cycle. In other words, the patrolmen work 2 hr in the tunnel at a time, 4 hr all told each day.

In the 3 years that we have been operating, physicians have been continually studying the *CO* condition, making blood analyses and watching every symptom of the patrolmen. As a matter of fact I believe the health of our patrolmen is probably a little better than the health of traffic patrolmen on the streets. The monotony of the job is the only complaint that we have about tunnel operation. One of the patrolmen made the statement he had counted all the tile in the tunnel in order to keep his mind off the monotony of watching the cars pass at 30 mph. He did not count them all but that is the way they feel about it, we have made certain moves to relieve the monotony, such as installing windshields in the tunnel which keeps the air current off the men. This air current in the tunnel is usually about 11 mph.

From the standpoint of the standard of 4 parts in 10,000, that is perfectly safe for people using the tunnel in the cars, but, as brought out, the smoke at times becomes excessive, especially on the upgrades where car and truck drivers shift gears. Cars will be running down-grade or along the level when they will hit an upgrade, without being aware of it until so much momentum has been lost that it is usually necessary to shift gears. This decrease in speed on the upgrade causes traffic to pile up some times so that carbon monoxide increases to a high figure.

As Dr. Sayers has said, we have 14 carbon monoxide recorders in the tunnel, one instrument for each duct, so that the supervisors can follow the traffic through the tunnel by looking at the carbon monoxide recorders. When the register shows 2½ parts in 10,000, the supervisor is permitted to observe this for a period of 5 min. At that time there is a bell that rings so that in case the supervisor happens to be busy, his attention is drawn to the *CO* condition. After 5 min if the *CO* does not diminish he puts in more air either by increasing the speed of a fan or operating additional fans.

On Sunday or holiday nights about 11:30 or 12:00 o'clock we usually have heavy traffic. We have carried as many as 2500 vehicles an hour. Ninety-eight per cent of them are passenger cars and we had some cases where the carbon monoxide has gone up to 4 and 4½ parts, but we are always able to hold it with our fan equipment. However, we are under contract whereby if we exceed a demand for a period of ½ hr we have to pay a monthly demand charge which follows through for a year. In order to save that demand charge we will allow the *CO* to go up a little over 2½ parts for a few minutes. So, if you look at our *CO* charts, you will note some high concentrations at times but at no time have we had any complaint from the public or employees on account of the *CO* conditions in the tunnel.

The smoke condition in the tunnel is one that we are studying, we have done some investigation work with the electric eye which is efficient in some ways but it still has many defects that prevent its use in vehicular tunnels for smoke detection. We are endeavoring to not only visualize the operation of the tunnel from the standpoint of carbon monoxide, but also atmospheric conditions, and I would say probably within the next year or so we will give our supervisors visualization of the absolute atmospheric conditions in the tunnel, and when we do that the operation of the tunnel will be practically perfect.

There seems to be but little change in the *CO* produced by automobiles today over that produced in 1919 when these tests were made. However, we have not made any further investigations along those lines. We have simply made analyses from our *CO* charts.

Mr. Posey states that he is building booths on his sidewalks. The reason is that, he is attempting to keep the patrolmen in the air for 8 hr. He will have to protect the men, if he is going to do that. We never expect to keep our patrolmen in the tunnel more than 2 hr at a time and we consider that safe, economical operation. The operating conditions of the two tunnels are very different. The tunnel operated by Mr. Posey is in two counties and it is a non-toll tunnel, whereas ours is a toll tunnel.

There are many things that I might have said in connection with the experimental work, but it had to be covered briefly. For instance, in the investigation that Professor Willard mentioned, it was discovered that there was a wide difference between the power required for blowing and exhausting.

We constructed a small test model, which was 1/10 the size of the Holland Tunnel duct and we got the idea that if we brought the air into this duct, which corresponds to the ceiling of the tunnel, at an angle it was the right thing to do, but the particular curvature of the exhaust ports was an accident and it happened this way: While we made 45 deg port openings we had a tinsmith out in Illinois make the little damper slides. He could not make sharp angles with such small ports, and as a result he made them with round edges. Fortunately, that was just exactly what we needed and they gave good results and we simply duplicated the port that he gave us.

Of course, if we could bring the air in at, say, 35 deg instead of 45 deg the power would be less; however, we were limited by the construction of the ceiling slab which was only 5 in. thick so we could not make the ports much less than 45 deg.

Just one more word about the design. Another reason that the system is as it is and not upside down from what we have, is on account of fires. The analysis of the two methods were really not overwhelmingly conclusive in favor of one over the other. In the conference that was held to determine which method should be adopted, it looked for a while as though the proposed method did not have many friends, but the final analysis was that we would adopt the method as proposed. In case of a fire if fresh air is put in at the top and the exhaust air taken out at the bottom all the smoke and all the heat is forced down on the occupants of the tunnel, also on the fireman. Whereas, if fresh air is put in at the bottom and taken out at the top, all the heat and the smoke is permitted to go up to the ceiling where it does not harm anybody and the fireman can get in underneath the smoke and the heat and extinguish the fire.

As a matter of fact, a fire in the tunnel does not cause as much excitement as a fire in the street. The smoke in the tunnel does not go any furtther than 30 ft on either side of the vehicle so that the patrolmen can get right next to the fire and put it out immediately. We average about 169 fires a year. Most of them are brake band fires.

In the time that we have operated, we have had no fear of any of the occupants of the tunnel from the standpoint of fire, which was one of the main points on which

the public wanted to be assured. They were afraid of carbon monoxide but they did not know much about it, but fire they did know about and every one asked the same question—What are you going to do in case of fire? We have absolutely assured the public that the tunnel is safe. We had a fire not long ago. Right back of the fire there was an automobile with women in it and they just sat there and enjoyed the show; no excitement at all. When women are quiet under those conditions, I think the ventilating system has proved its worth.

There are many things, as I said before, that I could talk to you about on this subject, but time does not permit. To give you some idea of the effectiveness of the ventilation as designed, the first year, 1928, we had a traffic of 8,745,000 cars. In 1929 we had 10,978,000 and in 1930 we had 12,067,000. Our maximum traffic day was about 56,597 cars. That was on May 11, 1930, and on that day we had, I believe about 3 stoppages of traffic.

No. 882

NOISE REDUCTION IN VENTILATING UNITS

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MEMBER

NOISE reduction is probably the most difficult and elusive problem in the unit ventilating industry. Although there is an immense fund of knowledge covering the theory of sound in general, there has been comparatively little research work done on the problem of reducing noise, the complexity of which is increased by the difficulty of evaluating the results of experiments in terms of the comparative disagreeableness of the noises produced. This paper includes a discussion of the problem as well as the results of some research which may be useful to those concerned with noise reduction in ventilating equipment.

The problem of noise reduction apparently involves a multitude of partial solutions. The first step is to prevent, as far as possible, the creation of noises. The next is to absorb or otherwise render inaudible that portion of the noise which could not be prevented.

SOURCES OF SOUND

Sound is created in unit ventilators from the following six major causes:

1. The mechanical vibrations of the rotating parts of the fans and the motor.
2. The hum in the transformer and motor windings.
3. The turbulence set up in the air during its passage through the unit.
4. The air vibration due to the action of the fan blades on the air.
5. The noise created by the mechanical movement of operating dampers.
6. Noises originating in the valves, radiator and traps.

Because of these several and separate sources, the resulting pulsation of the atmosphere is a most complex combination of widely varying intensities and wave frequencies. A large portion of the mechanical vibration of the rotating parts may be prevented by proper balancing, both statically and dynamically. Spring mountings between the motor and the motor base, minimum areas of contact between the motor and the ventilating machine and the

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use of various insulating materials will cause the vibration to be dissipated, although not before some portion of the vibration has been transferred to the air in the form of sound.

The second source of noise—motor and transformer hum—is caused by hysteresis and magnetic leakage. The quantity of noise from these sources is affected by the manner in which the windings are placed on the transformer core, and the manner in which the transformer is mounted. In general, a large air gap between the stator and the rotor will make a quiet motor, while a small air gap will produce greater noise. It may be necessary to sacrifice in some measure other desirable qualities such as power, compactness and efficiency, to obtain absence of noise.

There are critical velocities for all fluids below which the fluid will flow through a uniform duct without turbulence. Unfortunately these velocities are too low for most ventilating uses and it is therefore necessary to deal with air in a turbulent condition. There are, however, various degrees of turbulence, and the quantity of sound resulting from turbulence is a function of this degree. To prevent noise from turbulence, abrupt changes should be eliminated in area and direction, obstacles should be streamlined, and smooth surfaces should be provided.

The sound created by the action of the fan blades on the air cannot be entirely prevented by any means at present known. A low fan peripheral speed will reduce the noise, but only at the cost of reducing the possible air delivery from the fan. There has been some effort to prevent this noise by substituting a fan blade of curved streamline section for the present uniform thickness blade, although the results apparently have not been entirely satisfactory so far.

Dampers may be silenced by muffling the contact edges with felt, asbestos cloth or similar materials. Hinges and bearings should be such that comparative silence from these sources is assured in spite of lack of lubrication. Ordinary precautions as to proper draining will prevent any noises from condensate in the valves, traps and radiator. The radiator should have sufficient pitch, the amount of which depends on the cross sectional areas of the steam enclosure spaces and the steam velocities in these spaces.

SOUND ABSORPTION

Perhaps the best known method of reducing noise is by the absorption of the sound waves with some porous material. The interstices in the absorbing material are filled with air and present to the air passing over them, the open ends of a multitude of minute channels within which dissipation due to friction is very high. In order to prevent reflection an absorbing material must have approximately the same acoustic resistance as that of the adjacent air, and the pores and crevices must have a maximum of acoustical impedence and friction to extinguish the transmitted sound waves by conversion of the sound energy into heat. A dense, smooth-finish felt has a maximum of internal acoustic friction, but it has too much inertia, its acoustic resistance is too great, and therefore the sound waves are reflected rather than absorbed.

A series of tests were conducted to determine the proper density of felt to use as an absorbing material in unit ventilators. These tests indicated that felt was not suitable for this application. A large portion of the sound was

absorbed by coarse, loose felt, but only at a sacrifice of the rate of air delivery, due to the large amount of friction of the air stream in contact with the rough felt. The loss from friction was acceptably low with a smooth, dense felt, but the sound absorption was also low. A medium grade felt proved unsatisfactory, although this material is a useful absorbing material in many applications, particularly where the air velocities are low.

Felt, coated with a thin, smooth, flexible membrane has been suggested for the purpose of absorbing sound without interfering with the flow of the air. The purpose of the flexible skin is to present to the air a smooth, low friction surface which at the same time will be thin and flexible enough so that sound impulses will be transmitted through it to the felt, there to be absorbed. This combination proved ineffective because the reflection from such surfaces was too great, and consequently very little sound was absorbed.

There are other factors than the absorption coefficient to be weighed in the selection of an absorbing material. Among these are the space available, the cost, sanitation, durability and fire and water resistance. Sanitation should receive special consideration. Porous materials collect dirt readily and are difficult to clean, and unfortunately it is a characteristic of a good absorbing material that it be porous. The removal, by means of filters, of as large a portion as possible of the foreign matter from the air before it comes in contact with the sound absorbing material, is a partial solution. In considering the space available for an absorbing material, it has been found at the Riverbank Laboratories, Geneva, Illinois, that long narrow strips of an absorbing material are more effective per unit of exposed area than broad panels of the same material.

SELECTION OF MATERIALS

There is a wide variety of absorbing materials from which to choose, and some of these will not greatly impede the air flow. Perforated metal with 1/16 in. holes, 64 per square inch, when backed up by an absorbing material such as rock wool, 1 in. thick, has a coefficient of 0.84 at a frequency of 1,024, based on tests conducted by the U. S. Bureau of Standards. Laboratory tests have shown this to be a particularly good combination with little loss in volume of air delivered.

The absorption of sound has been measured by several methods and the absorption coefficients calculated for a great variety of materials. The absorption coefficients of materials tested by the U. S. Bureau of Standards and other laboratories are useful in the choice of an absorbing material.

The most effective thickness of absorbing material depends on the wave frequency of the sound. The absorption coefficients increase with the increase in thickness of material up to a point of maximum effectiveness which is attained at smaller thicknesses for the greater frequencies. For instance, in the higher ranges of 2,000 or over, almost no more absorbing power is obtained by increasing the thickness of felt over 3/8 in. In relation to the cost and to the space available, the law of diminishing returns becomes operative at some point less than the point of maximum absorbing power.

Mufflers, such as are used on automobiles, seem to have no practical application in unit ventilators. This type of silencer must necessarily occupy considerable space and must present to the air a prohibitive degree of air resistance.

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The arrangement and the design of baffles and dampers within the cabinets of unit ventilators has considerable effect on the noise output of these machines because these surfaces are apt to reflect and to focus the sound in an undesirable manner. If the machine discharge is directed vertically the noise is less noticeable than if it is directed toward the occupants of the room. Refraction of the sound, due to the difference in velocities and temperatures of the air currents in the room, changes, although to a minor degree, the direction in which the sound travels.

In the design of ventilating equipment it is often possible to fill in sharp corners and other turbulence-producing places with an absorbing material which will not only be effective in reducing noise but may actually reduce the total resistance to the flow of air. Recently a number of experimental machines were lined with one of the commercial absorbing materials. The material is composed of cedar excelsior which is hydraulically pressed into slabs while immersed in a magnesite cement. After baking, an extremely effective porous absorbing material is produced, which is permanent, fire proof and easy to apply.

The excelsior-magnesite material was placed in the machines in such a manner that it filled in several sharp corners. As a result, the air delivery of the machines was 2 per cent greater than before, and an appreciable reduction in the noise output was observed.

Sometimes canvas connections are placed between unit ventilators and duct work. This canvas yoke assists in reducing noise in two ways; it prevents the transmission of mechanical vibration between the machine and the duct work, and, also acts as an absorber.

The transmission of sound through the fan housings and the cabinets of unit ventilators is largely governed by the inertia and rigidity of these barriers. The amplitude of a sound vibration is very small, usually less than 0.005 in., depending on the loudness of the sound. If the walls have sufficient rigidity and inertia most of this movement will be prevented, and the transmission of sound through the walls thereby prevented.

SOUND INTERFERENCE

The phenomena of sound interference is of particular importance in unit ventilators. In most unit ventilators there are two fans assembled upon the shaft of a motor between them. These fans constitute two continuous sources of sound of the same amplitude and pitch, that is, the same frequency of alternation, pressure and rarefaction. Under these conditions at any instance there will be locations where the pressure from one source will exactly neutralize the partial vacuum from the second source, and therefore no sound will be audible. At one-half of a vibration later, these conditions will be reversed but the waves will still neutralize each other. At other and different points the two condensations may arrive simultaneously, at which point particularly loud sounds will be heard.

Due to the fact that there are several sources of sound in a unit ventilator and also that these sounds are of different pitch, the effect of interference is only partial, that is, there will never be complete neutralization at any point in a room of installation of one of these units, but there will be localities where the sound intensities will be changed by this interference and made

louder or softer. Interference is also to be expected from the reflection of the sound waves both in the unit itself and from the walls of the room of installation. The intereference between the two sound sources also has a reaction upon the sound output of those sources. This effect has been taken advantage of in the design of sound filters which prevent the emission of sound at those frequencies for which there is attenuation. No application has been made of such sound filters to unit ventilators so far as the author knows.

Resonance plays a part in the internal design of unit ventilators. Certain dampers and baffles are likely to be set in vibration by the sound waves and will then reinforce the frequency with which it is in tune.

The effect of reverberation inside the unit cabinets is practically negligible. It is proportional to the volume of the cabinet, and this is relatively small. The effect of reverberation in the room of installation, may, however, be great, although that is a matter which cannot be aided by any changes in the ventilating unit design.

Usually ventilating units are tested, and sometimes judged for noise, in the empty bare room of installation. This is not a fair test as it is not under these conditions that the machine will ordinarily be operated. The presence of an audience or of a class of students, with their large areas of sound absorbing clothing, will make an appreciable difference in the results of such a test.

CONCLUSIONS

The reduction of noise is a major problem in unit ventilation. If it were not for the restraining influence of noise, it would be possible to manufacture much more compact machines for delivering the same volume of air. Noise, therefore, has a direct and limiting effect on the entire design of these machines. Progress in noise reduction means progress in all phases of unit ventilator development.

DISCUSSION

S. A. CHALLMAN (WRITTEN): This paper shows a thorough and exhaustive study of the factors that enter into the manifestations of sound in ventilating units. The orderly presentation of these factors makes it an easy matter to follow the line of thought of the author. The validity of the conclusions which he reaches are so convincing that it would be useless to try to challenge any of them, even if one felt prone to do so.

Judging the ventilating units by the criteria set up by Mr. Ewald, it is quite evident that makers of these units have to an appreciable extent overcome objectionable noises. As a matter of fact, no sound of any kind is objectionable as long as it is not disconcerting to the person or persons trying to perform some definite task. There is such a thing as being hypercritical about the sound emanating from these units. A total absence of sound, if such were possible, would create an atmosphere of stagnation and would in the end prove just as annoying as any discordant sound. There is a hum of industry that is inseparably connected with any work that really counts and there is an application of mind to a problem which produces the best results only when this hum is present.

The ventilating equipment may not be the logical source from which this hum should emanate, but as long as it does not interfere with the work to be performed, no harm is done. We have for many years accepted as inconsequential the faint whistle of the air through the opening of the heat flue of a central fan system. It would seem that if we can tolerate the one we ought to be able to make some allow-

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ance for the other. The manufacturers of these units, it seems, have obviated the occasion for so doing.

JOHN HOWATT (WRITTEN): The engineering of sound insulation and noise elimination is receiving more and more attention each year because of the ever increasing demands for quietness. Noise is decidedly a commercial factor that must be considered in many cases. Floors and walls of hotels and apartment houses are sound proofed to provide privacy for the tenants, and in these days of acute competition for customers the salesman must be able to give assurance of quietness of rooms or quiet operation of the equipment he offers. Everywhere we may find instances where offices which would otherwise be desirable remain vacant because of excessive noise. We are now sound-isolation minded so the interesting paper by Mr. Ewald appears at an opportune time.

There is no class of equipment in which it is more essential that noises be eliminated than in unit ventilators. This equipment finds its greatest field of application in the ventilation of school classrooms. The machine is usually placed directly in the classroom with the teacher and pupils, so even slight noises can be heard above the usual noises of occupancy. A hum or vibration that might seem slight or negligible to one upon first entering the room may, after a time, so grate on the nerves of a distraught teacher that she will insist upon shutting down the equipment, choosing to get along without proper ventilation rather than longer tolerate the noise of the operating equipment. In fact, this is exactly what happens in many unit ventilator installations in classrooms. In my opinion, the noise of operation of unit ventilators is one of the major obstacles which the manufacturers of this class of equipment must overcome before they will be completely successful in their best field—that of school room ventilation. The units have been improved tremendously during the past 10 years and fill a real need as the thousands in satisfactory operation prove. When complete sound elimination is provided throughout the life of the equipment the remaining major criticism will have been met.

Almost any well constructed machine can be made to run quietly on test when new; witness how quietly the new automobile runs when the salesman is demonstrating it; but is it so quiet in operation two years later? The noise of improperly designed unit ventilators increases with years of use, especially those operating on alternating current because of wear and looseness of moving parts and the hum and vibration set up because of that looseness. So as Mr. Ewald's paper points out, it is not sufficient to take all possible precaution to prevent vibration in the machine, but means must also be provided to absorb any vibrations causing sound that might be set up. With the tendency of unit ventilator manufacturers to provide their equipment with condensor-transformer type motors, a long step has been taken to eliminate machine noises.

It is the opinion of the writer that quiet operation of such units over a long period of years can be obtained only by the use of large units operating at low speeds. It has been proved, however, that in order to produce proper distribution of air in the rooms a high velocity of discharge from the ventilator must be produced, and while high velocities present many problems from the standpoint of noise elimination, it is better for the manufacturers to face this problem and dispose of it rather than sacrifice any principle which will result in a decrease in the effectiveness of the unit as a ventilating device.

W. A. ROWE: I want to comment on the first portion of the paper having to do with the sources of sound. Those grouped under the first heading, "The Mechanical Vibration of Rotating Parts," there is little excuse for and the cure is obvious. The electrical noises are being taken care of to a greater extent every day by the motor manufacturers and where reduced to a minimum and are still objectionable can only be eliminated by a complete insulation of the metal parts of the motor from the building structure.

Most of these electrical vibrations, because of their high frequency, are most objectionable. They telegraph themselves throughout a building structure and become audible many times in the most remote parts of a building or in places where they are least expected. The only solution is an absolute insulation of the metal parts from the building structure. It is not sufficient to merely put a rubber pad under a motor, if bolts go through to the structure. It is not sufficient only to use rubber or felt or cork pads and rubber bushings around the bolts and washers between the heads and the nuts if the conduit leads to the motor from a piece of the structural framework of the building. The metal parts must be insulated.

There are two more items that to my mind are the frequent sources of noise in a ventilating unit. The first is under paragraph 8, namely the turbulence set up in the air during its passage through the unit. I am not speaking now of the unit ventilator so much as I am the ventilating fan. A modern, efficient ventilating fan operating within the range for which it is suitable, should not have turbulence within the unit. Turbulence, or pulsations which are called turbulence, are in almost every case in an efficient fan, due entirely to the failure to properly select the fan size for the system as installed. The *estimate* of the resistance of the system totally misrepresents the resistance of the system as it is actually put in, and over this factor the fan manufacturer has no control.

The setting of dampers, the putting into the system of obstructions which are of greater magnitude than assumed or estimated on the part of the engineer may result in a much higher resistance than is called for. If the system will not pass all of the air, it is discharged by the wheel, but because it cannot escape from the housing; it goes back into the wheel through the blades. As a result, the re-entry of air in any centrifugal wheel causes a rapid variation of pressure in the housing and that produces the pulsation which is so objectionable. The noise thus produced can be recognized immediately. It will be accompanied by weaving of the duct sides, no matter how stiff they are made. If ducts are made of boiler plate, you can put your hand on it and feel those pulsations. The greatest number of noise evils on fan ventilating installations are due solely to a failure to accurately determine the resistance of the system and to attach to it a fan of proper proportions.

This condition cannot be rectified by changing the speed of the fan. It is present at all speeds only its magnitude changes with the speeds. Regardless of the care exercised in developing methods for accurate measurement of noise in ventilating fans, all that effort will be voided where injudicious applications are made.

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No. 883

NOISE AND VENTILATION

By G. T. STANTON,[1] NEW YORK, N. Y.
(NON-MEMBER)

NOISE is variously defined, depending upon the viewpoint. The music from a radio, pleasant to the direct listener, is noise from the standpoint of a neighbor engaged in other pursuits. A popular definition, which seems adequately to express the usual conception, is that noise is any undesired sound.

The effects of noise on office employes has been studied extensively, and although it is difficult to evaluate the reduction of noise in terms of increased efficiency, greater output, higher quality or fewer errors, still the results, wherever tests have been carefully carried out, indicate that substantial dividends on the investment are possible. Noises, particularly when encountered during moments of relaxation, are detrimental in proportion to some psychological factor which may be termed their *annoyance*. Such annoyance is dependent upon many items and varies somewhat with each individual. For instance, to many people the monotonous repetition of a low intensity sound may create annoyance out of proportion to the loudness. However, the relationship between the annoyance of a sound and its pitch or frequency of vibration is rather definite. The annoyance or *stridency* of a sound is also associated with the degree to which its components lie in harmonic relationship to each other.

The architect and builder, to keep pace with modern demands, must, of necessity, consider the physiological aspect of his planning in order to avoid over-rapid obsolescence of the structures due to increased stringency of public standards for comfort and efficiency. In a building planned with these factors in mind, ventilation should, of course, be vital in making the structure suitable for continued efficient use, and in the future may be an ally of noise reduction. In metropolitan cities, the problem of noise is, to a large degree, that of overcoming the sounds from the street, from adjacent buildings and neighboring activities which stream through open windows. One solution of this problem is a ventilating system which, in addition to the many advantages obvious to ventilating engineers, affords the possibility of closing the windows and

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excluding the noise. However, when this advantageous factor in the employment of artificial ventilating and air-conditioning systems is considered, it is apparent that, above all, the ventilating system must not of itself create noise. Moreover, it must not act as a carrier of noise into the useful sections of the building.

The problem of reducing noise is broadly divided as follows:

1. Reduction of noise produced at the source.
2. Reduction of transmission of noise from the source to those areas under consideration.
3. Acoustic treatment in the area under consideration, to reduce the effect of the noise.

The problem as applied to ventilating systems appears to fall under items (1) and (2), namely, the reduction at the source and reduction in transmission. This also appears to be the most logical and satisfactory method in new construction. The architect, therefore, has an additional obligation imposed, that of securing in the finished building, conditions of quiet, suitable to the use of the property, judged by standards that are constantly increasing in severity. Hence, the noise problem is more and more frequently analyzed in the plan and specification stage.

The architect must have reasonably exact knowledge of the noise that will be created by such ventilating systems as may be installed, and include proper limits in the specifications. The performance of the system from a noise standpoint should be checked, for compliance, upon completion of the work. It would, therefore, seem desirable that more information be made available on this subject, that suitable measurements be made, and standards for allowable noise from different types of systems, be established.

NOISE MEASURING INSTRUMENTS

There are available two general methods of noise measurement, namely, *audiometric* and *instrumental,* which may be subdivided as follows:

1. Audiometric
   a. Overall masking or deafening effect
   b. Frequency band masking or deafening effect
   c. Beat frequency analysis
2. Instrumental
   a. Overall noise level
   b. Noise level by frequency bands
   c. Frequency and noise level analysis

In the audiometric methods of measurement, dependence is made on the aural faculties along with some device which will permit the assignment of a numerical magnitude to the noise effect. In the buzzer audiometer, the tone from an electric buzzer connected to a telephone receiver is adjusted in intensity until it is judged to be as loud as the noise or until it is masked by the noise. In the latter case, the external noise and the buzzer noise are impressed by means of an offset receiver cap on the air at the same time. The intensity of the test tone is varied by a calibrated attenuator until it can just barely be heard in the presence of noise. The difference of the reading obtained in the presence of the noise and that obtained in a very quiet place

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is a measure of the noise effect. In these and all threshold measurements the quantity measured is the deafening produced by the noise.

Methods utilizing the buzzer type audiometer give only the overall effect and give no indication of the quality or character of the noise. To obtain an indication of the noise quality, a variation of this audiometric method may be employed. In this case, the buzzer is replaced by equipment supplying either a series of constant frequency tones or warbling tones, each tone located in a different part of the frequency spectrum of audibility. The deafening effect of the noise is thus measured in each part of the sound spectrum and the noise audiogram can then be obtained. Fig. 1 indicates a typical warble frequency type of audiogram. Fig. 2 indicates a typical multiple tone type of audiogram.

In addition, by using a beat-frequency oscillator and audiometric methods, an approximate general analysis of the frequency spectrum of the noise can

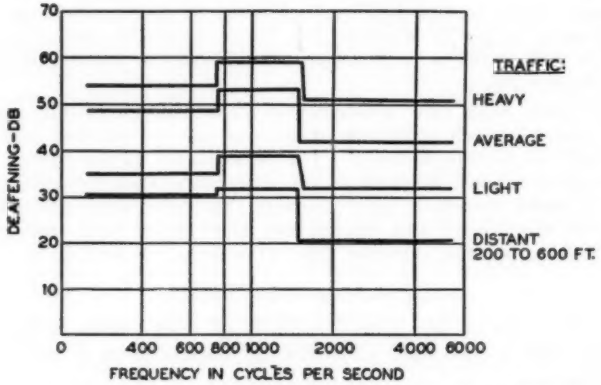

FIG. 1. AUDIOGRAMS OF OUTDOOR NOISE IN NEW YORK

be obtained. In this case beats between the tone of the oscillator swung through the audible range and the noise created by the source can be obtained. This method will give an indication of the magnitude of the components and it will give a general idea of the location of the more prominent components in the audible spectrum. Considerable experience and ingenuity are necessary to obtain satisfactory results from this method.

In the instrumental methods, all observations are entirely independent of the ear and yield purely physical measurements of the noise level. For overall measurements, a type of instrument generally called a noise meter may be used, which gives a direct reading of the noise level. The meter consists of a sensitive microphone, a calibrated attenuator, a frequency-weighting network, and a rectifying device with a direct reading meter. The ear is most sensitive in the region of 2,000 cycles but requires a great increase in intensity or energy flow going toward high and low ends of the audible frequency spectrum to produce an effect of equivalent loudness. The frequency-

weighting network is used to simulate the sensitivity of the human ear and thus obtain a measure of the loudness of the noise. Without this network the meter would indicate approximately the physical energy flow and would not indicate the physiological effect on hearing. The reading indicated by the noise meter is neither the true loudness level nor the true sensation units above the threshold of audibility but has arbitrarily been called the noise level. Experience has shown that the discrepancy caused by this assumption produces but a small error.

Since noise is a highly complex type of sound it is usually desirable to obtain information as to its character and quality. The use of band-pass filters

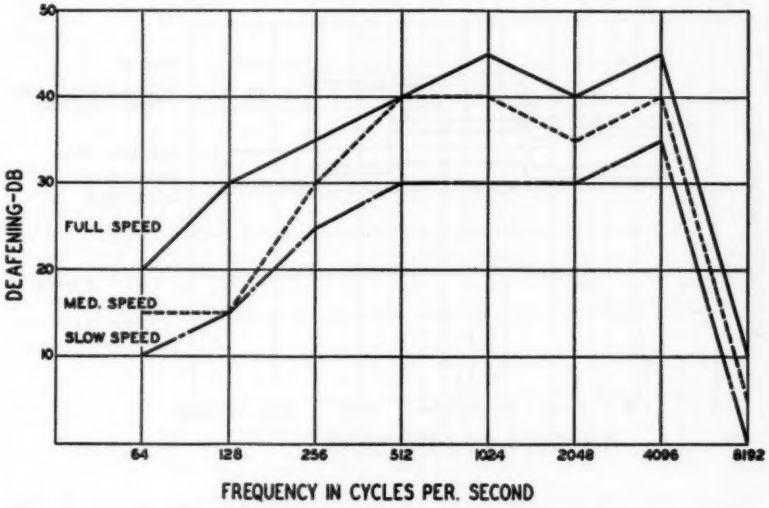

FIG. 2. AUDIOGRAM OF FAN AT VARIOUS SPEEDS

in conjunction with the noise meter will give a measure of the quality and distribution of the energy components in the audible spectrum. The method consists of taking measurements of the intensity in selected frequency bands, such as 0 to 500 cycles, then 500 to 1,500 cycles, etc. Thus a somewhat more detailed picture of the character of the noise being measured can be obtained and a comparison of the level of the different frequency bands made. Fig. 3 indicates typical noise-meter measurements utilizing band-pass filters. Such information is valuable in the comparative study of noise created by different sources. Though two sources of noise might give the same overall noise level, one noise source could be more annoying than the other. Measurements made using band-pass frequency analysis would probably show them to differ in their composition. It has been proved by experience that aural judgment of quality cannot be depended on in these circumstances except in unusual

and rare cases, due to the complexity of the frequency composition of normal noise sources.

For noise comparison, frequency band measurements are very satisfactory but for accurate analysis of the noise and its frequency components, it is necessary to decrease the width of the frequency bands. Harmonic analyzers have been devised to obtain automatically a frequency and level analysis of any sustained noise. This type of instrument takes rapid samplings of successive and very narrow frequency bands throughout the audible spectrum and records all readings. Such analyses give the complete history of the composition of the noise and allow easy segregation of the various factors contributing to the overall noise effect. Fig. 4 gives a typical harmonic analysis,

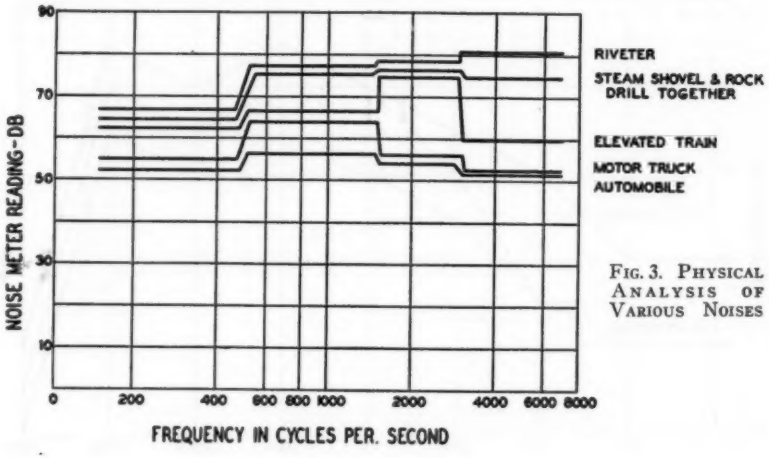

FIG. 3. PHYSICAL ANALYSIS OF VARIOUS NOISES

showing the various components and their magnitude. Information of this nature is of considerable importance to the designer of mechanical and electrical equipment, for it makes it possible to locate every noise source and thus apply corrective measures. In addition, such an harmonic analysis can serve as an excellent means of checking the progress of efforts intended to reduce the noise conditions created.

STANDARDIZATION OF NOISE MEASUREMENTS

In the study of the noise problems involved in the operation of any given ventilating equipment, a routine procedure of testing must be set up that will allow correlation of test results with those involved in direct practice. This is particularly necessary in checking the compliance with specifications on minimum generation of noise. In addition, it is inherently essential that results can be duplicated consistently so that control checks of performance can be obtained. Furthermore, if the data on any given system under test are to be

correlated with those obtained elsewhere on similar systems, the results must be reducible to a common basis.

It is not sufficient to make measurements of the respective noise levels. To have these measurements of value, the comparative conditions of tests must be taken into account. Control of the test conditions or means of calculating the equivalent acoustic conditions are required to obtain any comparative results. For example, consider a study of the noise created by a blower. Noise levels obtained with blowers set up on the test floor of the manufacturing shops will not be the same as measurements of noise levels obtained following

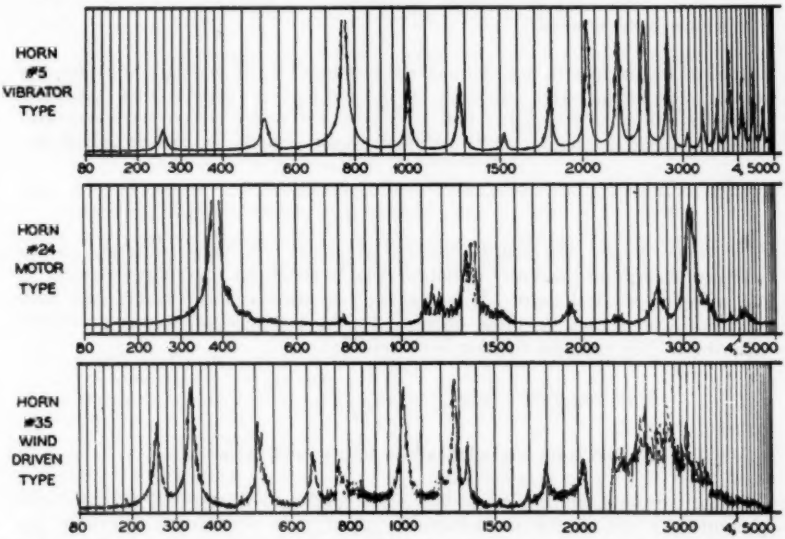

FIG. 4. TYPICAL HARMONIC ANALYSIS SHOWING VARIOUS COMPONENTS AND THEIR MAGNITUDE

the final installation of the blowers in their designated locations in theaters, office buildings, plants, etc. The difference of the conditions under which the tests are made are the controlling factors, provided it is assumed that the noise sources produce the same total acoustic energy output in the two cases.

There are numerous acoustic factors that can cause variation in the measurement of noise levels created by the same source in different locations. The variable factors primarily are as follows:

1. Cubic volume of rooms in which measurements are made
2. Total acoustic absorption present
3. The characteristic of the acoustic absorption over the audible spectrum
4. Presence of extraneous noise
5. Operating conditions of blower
6. Location at which measurement is made

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The introduction of these variable factors would appear to make it impossible to secure comparative results. However, by a few simple assumptions it is possible to utilize the general acoustic theory quite satisfactorily. For a steady state condition,

$$E = \frac{4E'}{av} \tag{1}$$

where

E = the energy of the noise per unit volume
E' = rate of noise energy emission by the source
a = the total acoustic absorption present
v = velocity of sound

It is of importance to note that all these units of noise energy are in physical units. For comparative purposes it is sufficient to assume that the noise energy emitted by the source will be a constant under all normal conditions. Then the relative factors for two different cases can be expressed as follows:

$$\frac{E_2}{E_1} = \frac{a_1}{a_2} \tag{2}$$

In other words, the energy density of noise will vary inversely as the acoustic absorption present. Equation (2) is based on the energy density of the noise in physical units. It is necessary to convert these values into units of noise level similar to those obtained by the instrumental means previously described. The simplest relation is as follows:

$$N_1 - N_2 = 10 \log_{10} \frac{E_1}{E_2} = 10 \log_{10} \frac{a_2}{a_1} \tag{3}$$

where

N = the indicated noise level obtained by instrumental means.

Thus, having a knowledge of the noise level and acoustic absorption units present at a test in the manufacturing plant it is possible to determine the noise level that might be expected following installation in a final location. In actual practice the relations shown will not necessarily hold entirely true due to the theoretical assumptions made. Correction should be made for the deviations in the assumption. The detailed methods of making these corrective calculations are not within the scope of this paper.

The fact that certain assumptions must be made makes it imperative that standard conditions be imposed for making all measurements of noise level. It is deemed highly desirable to make all measurements of noise in some selected test room arranged so that all variation of the acoustic factors be either eliminated or reduced to a minimum. In addition, it is absolutely necessary to maintain the normal performance of the equipment being tested so that no unsual factors be introduced that would not be finally present. By this is meant that a blower should be tested with a fixed length, or several selected lengths, if found desirable, of standard duct structure with normal static pressure and air delivery. Overall noise measurements can then be made which would include both the air and mechanical noise. Likewise, standard tests should be made and measurements obtained so that the only

introduction of noise into the measuring room can be through the duct system. This should more nearly simulate usual operating conditions, with a well-installed system in a well-constructed building. In this last case, studies of the air noise as introduced into the room can be made. In a ventilating system, movement of air by the blower above its critical velocity causes vortex motion and turbulent conditions which in turn is manifested by noise. Using standard test and measuring conditions, studies can be made of noise conditions with respect to the static pressure, speed of the blower, duct section, material of duct structure and other determining factors which will generally be comparable.

By using some standard test procedure for measuring noise, the following benefits are obtainable.

1. Shop measurements may be converted into performance ratings in any given location.
2. A basis of rating equipment on noise performance may be established.
3. Standards of minimum allowable noise for different types of systems, relative to air delivery, etc., may be established.
4. All data taken will be capable of correlation and comparison, permitting more rapid accumulation of empirical data.

All of these seem necessary in view of the increasing interest in the subject of noise, and the demand for consideration of this factor in the design of many modern structures.

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DISCUSSION

ALBERT BUENGER (WRITTEN): Mr. Stanton in his paper suggests one phase of the standardization of noise measurements which, in my opinion, should be stressed, and that is the establishment of a *standard of quietness* for ventilating equipment. As often as not a motor selected from the floor in the shop is sent into the field as *quiet* and when installed proves to be annoying. It also happens that the motor or fan is quiet and the noise comes from faulty duct work although the equipment is blamed.

If it were possible to send out equipment with a *noise rating* just as equipment is rated for capacity, speed and power, the manufacturers of the equipment as well as the purchasers would be benefited. I believe that the time is not very far distant when this can be accomplished.

J. C. STEINBERG (WRITTEN): This paper is a concise and readable account of the bearing of noise and its measurement on ventilating systems. As the author points out, the problem of reducing noise involves, in part, a reduction of the transmission of noise from the source to those areas under consideration. In many cases,

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the air ducts are the avenues that are chiefly responsible for the transmission of noise, in other words, they are the transmission lines. This noise arises from two sources, (1) from the unit which supplies the air, and (2) from the frictional modulation of the direct air stream as it passes along the duct.

Thus the supply unit furnishes both direct and alternating current to the line, the direct current being the air stream, and the alternating current the undesired noise. An ideal air duct would be a low pass acoustic filter with a cut-off frequency below the lowest audible pitch. Such a filter would freely pass a direct air stream but would attenuate audible noise.

The noise which arises from frictional modulation cannot be eliminated by the filtering action of the duct since it arises in the elements of the filter. It can be reduced by stream line ducts, by absorption, and by a reduction in velocity.

The amount of reduction needed is determined by the level of noise, due to other sources, in the area under consideration. To be unnoticed, the ventilating noise should be 15 decibels or more below the room noise level.

It seems that the cooperative efforts of the architect, the ventilating and acoustic engineer could do much to incorporate desirable filtering and absorbing properties into air ducts.

MR. STANTON: I noticed particularly the demand for a specification for acceptable noise levels from the different types of systems in rooms intended for different uses. I believe something of that sort might be possible specifying, of course, for average conditions. However, the most stringent cases into which you will have to put air conditioning or ventilating systems will be those where the management has determined from physiological studies or something of that kind what the requirements may be. I have in mind one specific case of a private hospital which is constructing a wing for the insane consisting of private suites of rooms. Room and bath, and room, bath and sun parlor, obviously intended for rather wealthy patients, and in which it is absolutely necessary in the treatment that any disturbance which may occur in one of the rooms be inaudible in the other rooms.

Obviously, that means that the rooms must be sealed up from outdoors, using double windows, double doors and double walls with air conditioning a necessity. In that case it is not entirely a question of the noise created in the system but it is a question of the noise transmission through the system and the standards there are obviously rigorous.

Dr. Steinberg mentioned that it would be necessary, if the system noise were to be inaudible, that it be approximately 15 decibels below the room noise level. In the comments from the floor it was stated that where a room noise level below 15 decibels existed, in all probability the system would be audible. However, room noise levels below 15 decibels above threshold are very special conditions and in general exist only in the most carefully constructed studies or laboratories.

No. 884

SOME STUDIES ON THE ABSORPTION OF NOISE IN VENTILATING DUCTS

By G. L. LARSON[1] *(MEMBER),* AND R. F. NORRIS,[2] *(NON-MEMBER),* MADISON, WIS.

IN THE ventilating or conditioning of schools, theaters, churches, assembly halls, offices, recording studios, broadcasting studios, etc., quiet operation is essential and must be accomplished if the ventilating plant is to be considered a complete success.

The designer of the future must give more consideration to this factor than has been given in the past. The United States is becoming a noise conscious nation and as more facts about noise and its effects become known, the demand for noiseless ventilating and air conditioning systems will increase.

Many otherwise successful ventilating systems are considered failures because they create too much noise. Recently in one of the new and best ventilated churches the minister opened his talk with the remark: "Will someone please shut down the ventilating system so that I may begin my sermon."

A study of ventilating system noises is essential to successful design not only on account of the effect of noise upon human beings but also on account of the limitation that noise puts on the maximum velocities that can be used in duct design.

In buildings where space is at a premium, it is very desirable to be able to reduce the size of ventilating ducts by using velocities that are higher than those now considered good practice. The present limitations to higher velocities are (a) noise, and (b) cost of power required to produce high velocities. If noise can be sufficiently reduced by proper acoustical treatment of the ducts, then the only limitation to higher velocities is the cost of power required to produce these higher velocities.

In any given problem the value of the space saved can be balanced against the cost of the acoustical treatment plus the cost of increased power, but the very valuable item of quiet operation is hard to evaluate in dollars and cents.

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[2] Research Engineer, Department of Sound Control, C. F. Burgess Laboratories, Inc.

Presented at the 37th Annual Meeting of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, Pittsburgh, Pa., January, 1931, by R. F. Norris.

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ACOUSTICAL DISCUSSION

There has been so much said in literature during the last few years about the harmfulness and wastefulness of noise, that it does not seem necessary to discuss that phase of the problem in this paper. However, a resumé of the physical nature and the causes of noise may help to clarify the remainder of this paper. Noise, such as the ear distinguishes, is a phenomenon which occurs in the surrounding air. Vibrating bodies set up pulsations in the air which cause alternating rarefactions and condensations. These proceed from the vibrating body at a velocity which depends to a slight degree on the temperature and is approximately 1,180 ft per second at 70 F. These changes in pressure are very slight in magnitude, but act upon the ear drum in such a way as to stimulate the auditory nerves which in turn convey the impression of noise to the brain. Noise in short, then, is a vibration set up in the air by some vibrating medium.

In this article, sound and noise will be considered as synonymous. From the æsthetic standpoint, music for instance would not be considered noise. However, even the best of music can be very disturbing under certain circumstances.

In ventilating work there are three major sources of noise. One is the noise generated by the fan blades passing the lip of the fan housing. The blades will cause a fluctuation in air pressure at this point which will correspond in frequency to the number of times per second that a fan blade passes the lip. In multi-blade high-speed fans this frequency can reach rather high values. The second source of noise is that generated by the motor which drives the fan. A slight eccentricity of the motor shaft will cause a wobble which will set up vibration in the machine and in the foundation. Even in the best of bearings there may be inaccuracies which will generate noise. If an alternating current motor is used the field coils may set up a hum. The third source of noise is the rush of the air through the ducts. The air going through the ducts and around bends will set up eddy currents in itself, which will produce noise and high frequency sound vibrations. This is particularly true if high velocities and small ducts are used.

Noise may be transmitted from the fan equipment in two ways. It may be transmitted through the air stream itself or as a mechanical vibration in the duct. If it is transmitted as a mechanical vibration of the duct and remains a mechanical vibration to the end of the duct, it will not cause a sensation of sound in the ears of persons occupying the rooms which the ducts serve. However, there are usually rather large areas in the ducts which can and will vibrate in response to this mechanical vibration, thus coupling it into the air at various points along the length of the duct. After the vibration has been coupled to the air it becomes audible and is transmitted through the air stream in the same way as are the other air vibrations. All of these noises may be transmitted through the ventilating system since the duct itself acts as an excellent speaking tube from the fan to the rooms which it serves. The mechanical vibration, due to the motor and the vibration of the fan, may be kept out of the duct by using a flexible connection between the fan housing and the beginning of the duct system. The noise produced by the blades of the fan passing the lip of the fan housing will be generated directly in the

air stream and will be transmitted very efficiently to all of the rooms as will be the noise generated by the air itself, due to its speed through the ducts and around bends in the ducts.

The idea of measuring sound is rather new but certain units of measurement have been established. A sound wave in the air may have the following characteristics:

A. Frequency. The sound impulses may follow each other more or less rapidly. If the impulses follow closely, one behind the other, the frequency or pitch of the sound will be high. If the impulses are spread out so that each follows its neighbor by greater time intervals, the pitch of the sound will be low. A person having normal hearing can hear sounds ranging from 32 impulses per second to 15,000 impulses per second. This range of frequency is termed the audible range.

B. Amplitude. A sound wave of any one frequency may differ from another sound wave of the same frequency by having a different amplitude. In other words, the maximum variation of pressure of one sound wave will be greater than the maximum variation of pressure in the other. The sound wave having the greater variation in pressure will sound louder to the ear. In measuring sound the amplitude of the sound wave is not usually measured as such, but measurement is made of the intensity of sound, which is the total energy of sound. The intensity ratio of the loudest sound which the ear can stand to the softest sound that the ear can recognize as such, is greater than one billion to one. The human ear does not respond directly to the intensity of the sound but rather responds to the logarithm of intensity. For this reason it was necessary to develop a unit of loudness. This unit has been called the *decibel.* Mathematically the decibel may be expressed as

$$10 \log_{10} \frac{I_1}{I_0}$$

Where I_1 and I_0 are two intensities, I_0 being the intensity from which the loudness of the sound is measured. In acoustical measurements, I_0 is usually taken as that intensity which is just barely audible in an absolutely quiet room. This value is called the threshold of hearing and is represented by zero on the decibel scale.

Physically an increase or decrease of loudness of one decibel is the smallest increase or decrease that can be detected by the ear. Consquently, if one sound is louder by one decibel than another it will be just noticeably louder. Ten decibels increase in loudness represents an intensity increase of 10 to 1, 20 decibels increase in loudness represents an intensity increase of 100 to 1, and 30 decibels increase in loudness represents an intensity increase of 1,000 to 1, etc.

Fig. 1 shows the loudness of some familiar noises on the decibel scale.

OBJECT OF TESTS

The object of the tests reported in this paper was to determine the effect upon sound intensity of various arrangements of acoustically lined and unlined ventilating ducts and the effect of such lining upon resistance to air flow at air velocities varying from 500 to 5,000 ft. per minute.

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DESCRIPTION OF TEST APPARATUS

The 30-ft. duct which was used for these tests was made up of interchangeable 2-ft. sections. Each section was 10 by 10 in. on the inside. This permitted various combinations ranging from all lined to all unlined ducts. The

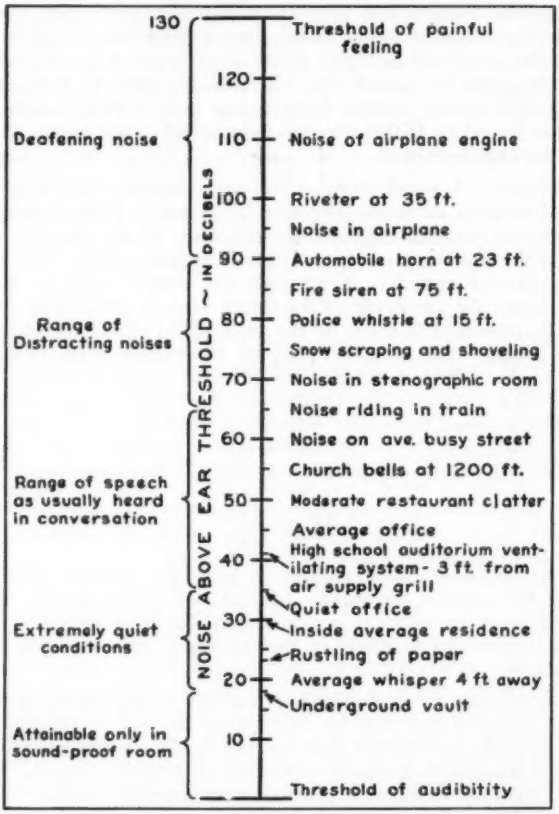

FIG. 1. LEVEL OF NOISES FROM FAMILIAR SOURCES

test duct is shown in Fig. 2 with 9 sections (18 ft.) of unlined duct at the intake end and 6 sections (12 ft.) of lined duct at the discharge end. One of each of the two types of section is shown in Fig. 3. The duct metal was galvanized and No. 26-gage thickness. Each 2-ft. section of unlined duct had a stiffener rib at one end. The lined sections had an enlarged casing on the outside 12 by 12 in. This provided a cavity one inch thick into which a 1-in. layer of

FIG. 2. THIRTY FEET OF TEST DUCT EXTENDING FROM SOUND PROOF PARTITION

wood fibre insulating blanket was laid. This blanket was covered on the inside with perforated metal which had 32 holes per square inch. These holes were 0.075 in. in diameter and were made by punching. Another scheme of lining was used whereby the wood fibre insulating blanket was held in the duct by four ¾ by ¾-in. angles cut from the perforated metal.

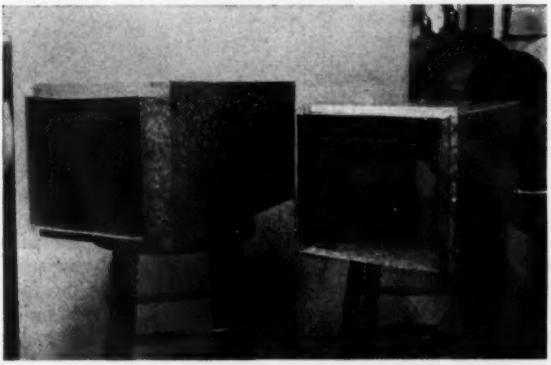

FIG. 3. CONSTRUCTION OF TEST DUCT SECTIONS

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In order to make certain that the measured ventilating noises were all coming through or from the duct, the blower and motor drive were completely boxed in with sound insulating material, as shown in Fig. 4. The fan was connected to a 12-in. circular metal duct by means of rubber lined canvas. After a 3-ft. radius right angle turn, there was 10 ft. of straight pipe before a

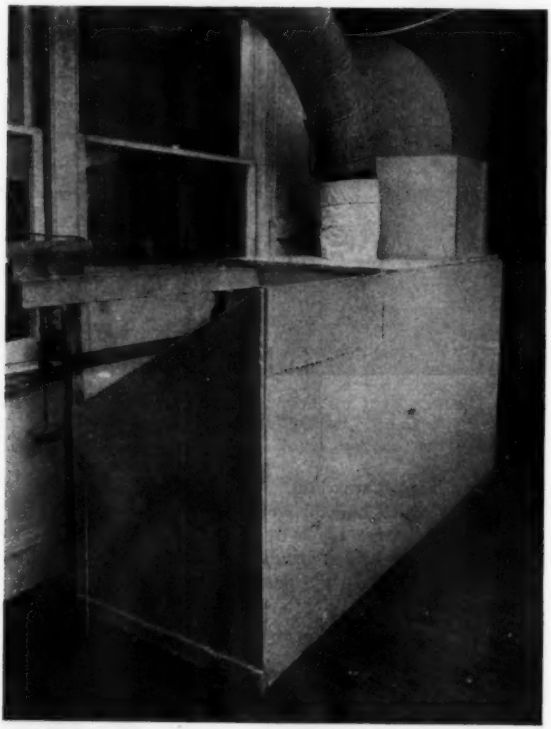

FIG. 4. ACOUSTICAL HOUSING OF FAN AND DYNAMOMETER SHOWING SLEEVE CONNECTION TO DUCT

Pitot tube station. Beyond this station, there was 4 ft. of circular pipe followed by 3-ft. adapter section which connected the 12-in. round duct to the 10-in. square test duct. Between the Pitot tube station and the adapter section, a sound partition was constructed. This wall was heavily covered on the fan side with acoustical insulation, and it completely separated the fan room from the room in which the test duct was located. A part of this wall is shown in the background of Fig. 2. All windows and doors to both rooms were kept closed throughout all tests.

The brake horsepower required by the fan motor was measured by a scale

beam connected to the dynamometer mounting of the motor frame. The beam is shown in Fig. 4. The fan speed was determined by means of a calibrated electric tachometer. The air flow was measured by means of a Pitot tube and Wahlen gage. The Pitot tube was of standard construction. The Wahlen gage is a very sensitive differential gage capable of measuring pressures up to one inch with an accuracy of 0.001 in. of alcohol.

The measurement of noise was accomplished by means of a microphone and acoustimeter set up as shown in Fig. 5. The microphone was placed 2 ft.

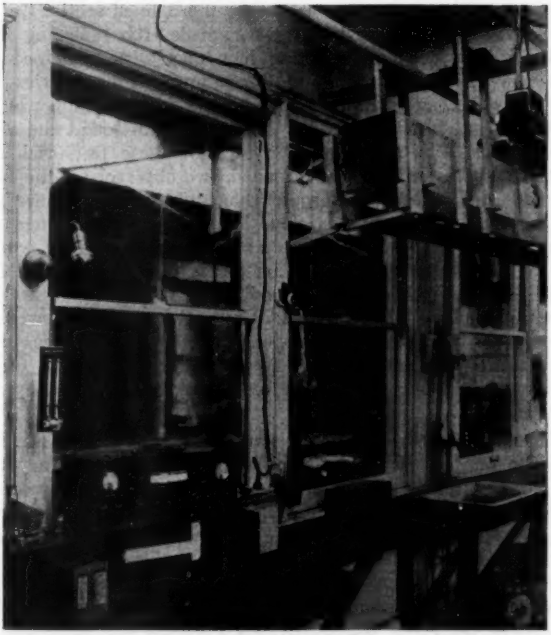

FIG. 5. ARRANGEMENT OF MICROPHONE AND ACOUSTIMETER FOR MEASURING NOISE LEVELS

from the center of the test duct at right angles to and in line with the outlet end. Both the microphone and the electrical cord connecting it to the acoustimeter were supported in such a manner as to avoid any foreign vibrations which might affect the acoustimeter readings.

The acoustimeter is comprised of a condenser microphone, a special amplifier and a thermal ammeter. Sound waves striking the microphone generate an electric wave which is proportional to the amplitude of the sound wave. This electric wave is amplified and then impressed on the heater of a vacuum thermal couple. The thermal couple generates a current which is proportional to the square of the impressed wave. This current is measured by means of a very

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sensitive micro-ammeter. Since the intensity of the sound wave is proportional to the square of its amplitude, the reading of the micro-ammeter is directly proportional to the intensity of the sound.

METHOD OF PROCEDURE

The first test was made with the entire duct built up from sections which were lined with both the acoustical material and the perforated metal. Each test consisted of four runs at fan speeds of 200, 500, 800, and 1,100 rpm to furnish a wide range of air velocities.

The second test was made after replacing the 3 lined sections at the outlet end with unlined sections. The third test was made after replacing 3 more lined sections with unlined sections, resulting in a duct with 6 unlined sections at outlet. This procedure was followed throughout the first series of 6 tests. This series covered a range from all lined duct to all unlined duct in increments of 6 ft. During each run for each of the 6 tests, both sound and air measurements were taken. The procedure for each of these tests was as follows. After the speed of the fan had been adjusted, two observers in the fan chamber observed speed of fan, load on torque scale, differential pressure across fan, and static pressure at Pitot tube station. The Pitot-tube readings were taken at 20 points in the 12-in. diameter round pipe. The readings were in thousandths of an inch of alcohol as read on the Wahlen gage. The 20 observation points in the duct included a horizontal and a vertical traverse with definite points located mathematically so that 4 values of velocity head were taken for each of five equal areas of cross section. From these readings and the air density, the average velocity head was determined. A third observer stationed at the acoustimeter in the duct room took a series of observations of the noise intensity.

Each of the first 6 tests showed the same air flow and the same horsepower required by the fan. Since the tests included 6 combinations varying from all lined to all unlined duct, the air was not measured for the succeeding tests involving other combinations of the same duct sections. Thus several tests were made by adjusting the speed of the fan and observing the noise intensity. Five such tests obtained data for combinations where the amount of lining at the outlet was varied in increments of 3 sections of duct (6 ft.). The various duct arrangements for each test are shown by Fig. 6.

After all tests involving perforated metal lining had been completed, the perforated metal was removed from the lined ducts and the acoustical material held in by four perforated metal angles placed at the corners. During the tests involving this type of lining, both air and noise measurement observations were taken.

For the higher velocities it was a simple matter to measure the noise intensity; but for the low velocities, and especially the series involving lined duct, foreign noises interfered. For this reason, it was necessary to make these tests at night when more quiet conditions could be obtained. Tests were repeated until an adequate check upon all results was made.

DISCUSSION OF RESULTS

Fig. 7 shows the curves pertaining to air movement. The solid line curves were obtained from data observed during the first 6 tests. There was no

difference between the air flow or horsepower input with unlined duct and the duct which had the perforated metal on the inside. This was not the case where the acoustical material was exposed. The duct resistance increased and

Each Section 2'-0" long and 10"×10" inside.
U = Unlined duct.
M = Acoustically lined duct- perforated metal and wood fibre blanket.
L = Acoustically lined duct- wood fibre blanket.

TEST NO.	SECTION NUMBER ←DISCHARGE										← INTAKE				
	1	2	3	4	5	6	7	8	9	10	11	12	13	14	15
1	M	M	M	M	M	M	M	M	M	M	M	M	M	M	M
2	U	U	U	M	M	M	M	M	M	M	M	M	M	M	M
3	U	U	U	U	U	U	M	M	M	M	M	M	M	M	M
4	U	U	U	U	U	U	U	U	U	M	M	M	M	M	M
5	U	U	U	U	U	U	U	U	U	U	U	U	M	M	M
6	U	U	U	U	U	U	U	U	U	U	U	U	U	U	U
7	M	M	M	U	U	U	U	U	U	U	U	U	U	U	U
8	M	M	M	M	M	M	U	U	U	U	U	U	U	U	U
9	M	M	M	M	M	M	M	M	M	U	U	U	U	U	U
10	M	M	M	M	M	M	M	M	M	M	M	M	U	U	U
11	M	M	M	M	M	M	M	M	M	M	M	M	M	M	M
12	U	U	U	U	U	U	U	U	U	U	U	U	U	U	U
13	M	M	M	U	U	U	U	U	U	U	U	U	M	M	M
14	M	U	M	U	M	U	M	U	M	U	M	U	M	U	M
15	L	L	L	L	L	L	L	L	L	L	L	L	L	L	L
16	U	U	U	U	U	U	U	U	U	U	U	U	M	M	M
17	M	M	M	U	U	U	U	U	U	U	U	U	U	U	U
18	U	U	U	U	U	U	L	L	L	L	L	L	L	L	L

FIG. 6. OUTLINE OF TESTS AND DUCT ARRANGEMENTS

this is shown by the broken line curves. The capacity of the fan for any given speed was reduced and this resulted in a lower horsepower input.

Fig. 8 shows how the noises from the various duct arrangements compare over a range of air velocities. Points are shown for only 4 of the curves and curves for 3 of the tests were not drawn in order to avoid overcrowding the curve sheet. There was not a great deal of difference in the noise intensity for several of the tests. A matter of 1 decibel is very difficult to measure

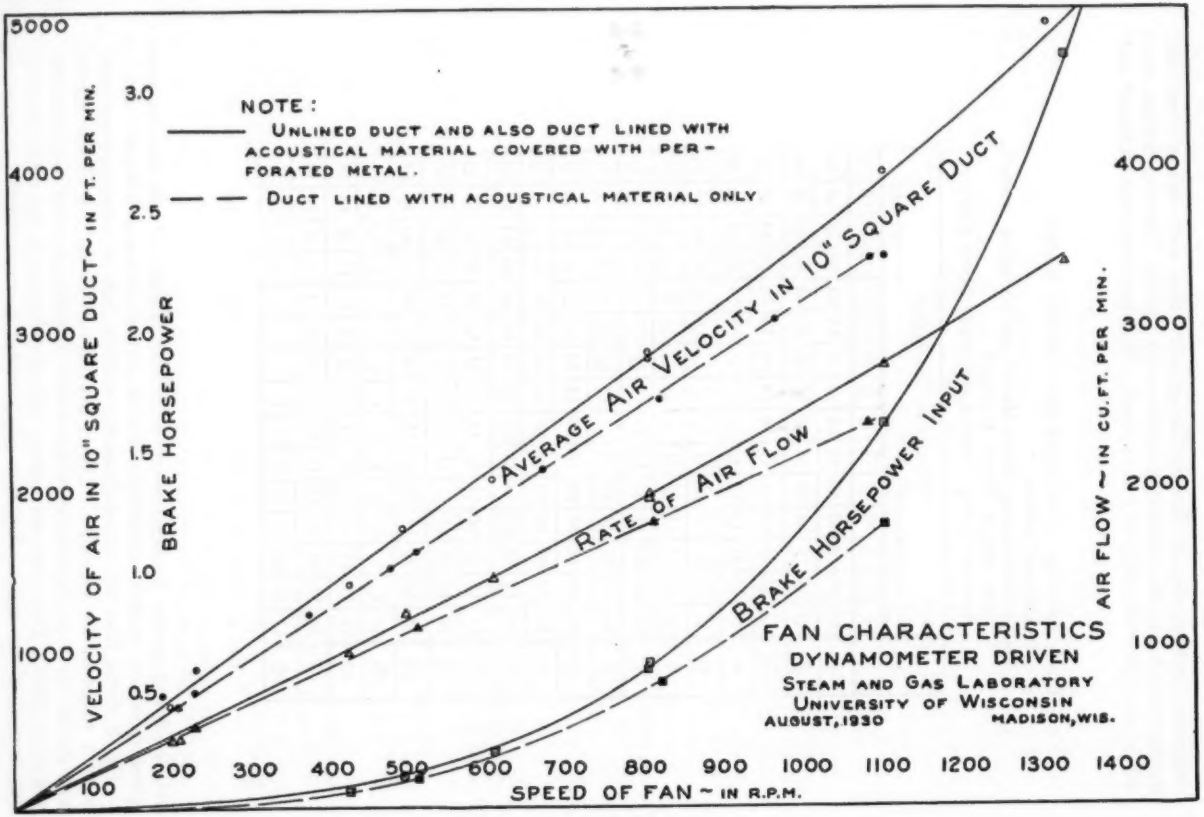
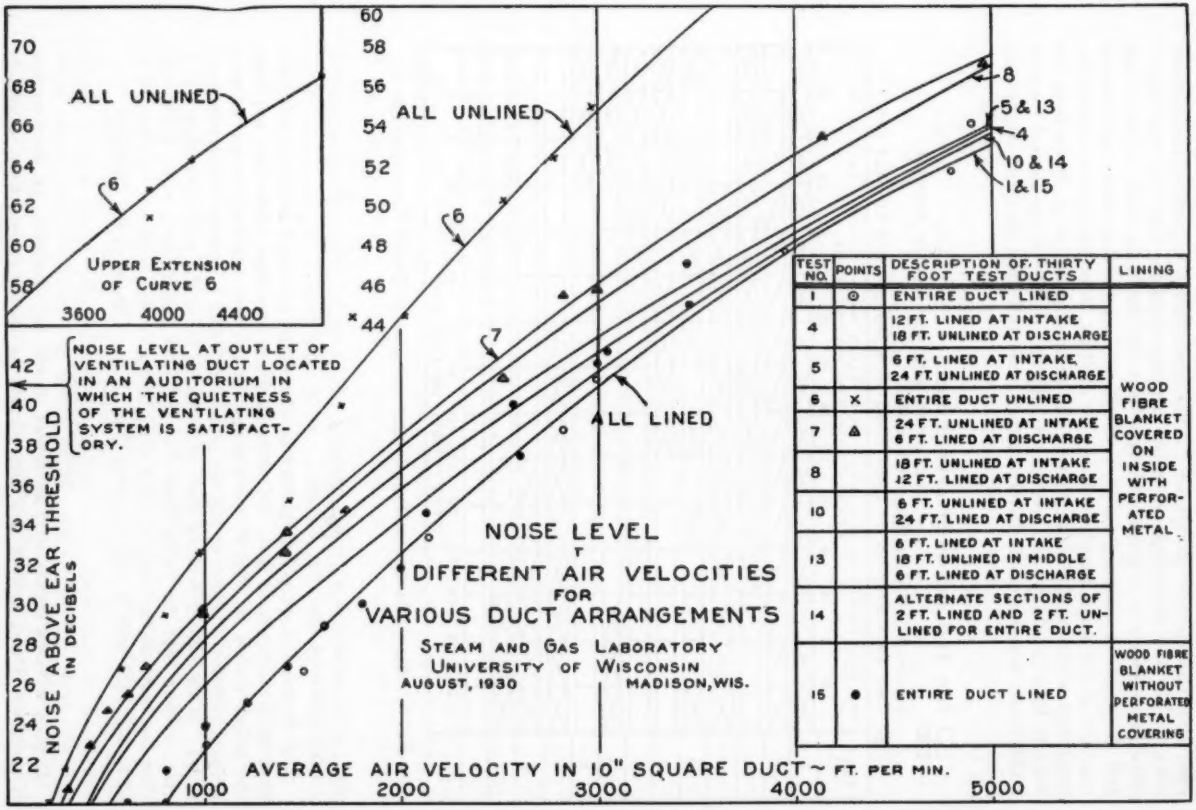


FIG. 7. AIR FLOW CURVES

FIG. 7. AIR FLOW CURVES



TEST NO.	POINTS	DESCRIPTION OF THIRTY FOOT TEST DUCTS	LINING
1	⊙	ENTIRE DUCT LINED	WOOD FIBRE BLANKET COVERED ON INSIDE WITH PERFORATED METAL
4		12 FT. LINED AT INTAKE 18 FT. UNLINED AT DISCHARGE	
5		6 FT. LINED AT INTAKE 24 FT. UNLINED AT DISCHARGE	
6	×	ENTIRE DUCT UNLINED	
7	△	24 FT. UNLINED AT INTAKE 6 FT. LINED AT DISCHARGE	
8		18 FT. UNLINED AT INTAKE 12 FT. LINED AT DISCHARGE	
10		6 FT. UNLINED AT INTAKE 24 FT. LINED AT DISCHARGE	
13		6 FT. LINED AT INTAKE 18 FT. UNLINED IN MIDDLE 6 FT. LINED AT DISCHARGE	
14		ALTERNATE SECTIONS OF 2 FT. LINED AND 2 FT. UNLINED FOR ENTIRE DUCT.	
15	●	ENTIRE DUCT LINED	WOOD FIBRE BLANKET WITHOUT PERFORATED METAL COVERING

FIG. 8. NOISE LEVEL CURVES

even with the sensitive acoustimeter. The noise increases with increase of air velocity but the increment of noise increase becomes less for the higher velocities. There is a great difference between the noise from a lined duct and that from an unlined duct. For example, if the two ducts were to be operated at the same sound level, say 40 decibels, then the air velocity in the lined duct could be 1.8 times that in the unlined duct, or at 26 decibels the velocity ratio would be 2.5. At an air velocity of 800 ft. per minute there is a difference in noise of 10 decibels between the full lined duct and the unlined

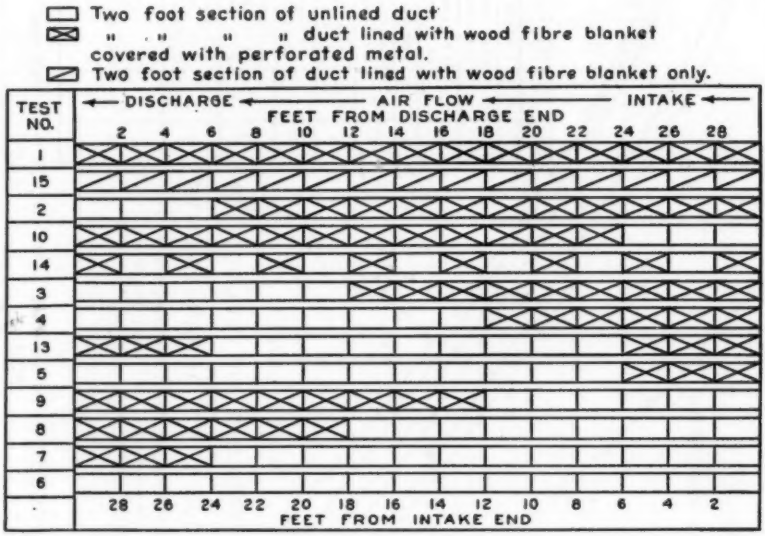


FIG. 9. ARRANGEMENT OF TEST DUCTS IN ORDER OF THEIR EFFECTIVENESS

duct, or a reduction of noise intensity of 10 to 1. Other relations can be readily obtained from the curves.

Fig. 9 shows a graphical picture of the ducts arranged in order with the most quiet arrangement first. Of course, the full lined duct ranks first and from the points shown in Fig. 8, there seems to be very little difference from a noise standpoint, whether or not the acoustical lining is covered with perforated metal. What little difference there is seems to favor the metal covering. It is quite evident that the lining placed at the inlet results in a more quiet duct than one that has the same or even more lining at the discharge. In fact, 6 ft. of lining at the inlet is slightly better than 18 ft. of lining at the discharge.

CONCLUSIONS

It goes without saying that all noises should be eliminated at their source, if possible. Quiet motors and fans should be specified, and they should be

mounted on special foundations designed for absorbing vibrations. Fans should be connected to the duct system by means of canvas connections at both inlet and outlet of fan. Endless belts should be used where motors are not direct connected to fans. In special cases, it may be necessary to treat the fan room with acoustic material.

But even if all these precautions are taken, considerable noise may be carried by the duct system from the fan room to the rooms served, or disagreeable noises may be developed in the duct system itself, particularly in systems using high air velocities. The tests show that these noises can be greatly reduced by lining the ducts with sound absorbing material. As would be expected the best results are obtained when the entire duct is lined.

When lining the entire duct is not practical, it is more economical to line at the inlet than it is to line at the outlet. The tests show that 6 ft. of lining at the inlet is more effective than 18 ft. of lining at the outlet. However, it may be necessary to place some lining near the outlet of long ducts in order to absorb noises developed in the ducts themselves.

Acoustic material with a perforated metal surface does not increase the resistance to air flow above that in an unlined metal duct. The same acoustic material without the perforated metal surface has about the same sound absorbing capacity as the other, but it has a greater resistance to air flow. Furthermore, it may disintegrate and be carried away by the air when used with high velocities.

Acoustic material applied to ventilating ducts makes it possible to use higher velocities without getting into the range of disagreeable noises.

DISCUSSION

ALBERT BUENGER (WRITTEN): It is extremely gratifying and interesting to note that some papers on the subject of noise in ventilating systems are to be presented at this meeting. Without a doubt many otherwise excellent ventilating systems have been ruined by noise.

The thought has often occurred to me, especially when confronted with the problem of getting a large duct in a restricted space whether much higher velocities could not efficiently be used. The main drawback that has presented itself, as the authors state, is the noise. I believe that Professor Larson and Mr. Norris have laid a foundation for an era of higher velocities in ventilation work.

I should also like to see these experiments either carried out, or further evaluated, so that some practical standards for sound insulation can be set up. A formula or relationship similar to Dr. Watson's formula which gives the amount of acoustic absorption required for any given room to obtain certain periods of reverberation. A formula such as this might express a relationship between the area and length of the duct, the air velocity, the distance of the fan from the first outlet to the fan, and the amount of sound absorbing material required in the duct to obtain a certain accepted loudness expressed in sensation units or decibles at the register outlet.

To illustrate the need of this I will relate an example from my own meager experience in this particular field: We are just completing a small auditorium in the two upper stores of the Lowry Medical Arts Building in St. Paul. The officers of the *Medical Society* who will use this auditorium for their meetings and lectures were anxious that the ventilating system be as quiet as possible, so as not to be annoying during a lecture. In order to insure this the fans and motors were selected for quiet operation, low velocities in the ducts were adhered to, the fans were set on

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insulating foundations, and the main trench ducts were lined with ½ in. of wood fibre blanket from the fan to the point where the branch ducts were taken off; each of the branches is isolated with a canvas connection from the main. Although final tests of this system have not been made, the preliminary tests show that the desired results have been achieved.

Here is a case where we were in the dark as to what acoustic absorption was required to get a minimum loudness at the outlets of the duct system. If a formula as suggested in the foregoing were available, our problem would have been simplified.

It is the experiments which this paper discusses which will give us the necessary fundamental information so that the desired results can be obtained surely and in the most economical manner.

S. I. ROTTMAYER (WRITTEN): Sound absorbing materials may be rated for absorption of sound of a certain pitch. Thus for ventilating work, a pitch of 512 vibrations per second (one octave above middle C) is commonly used, and a given material may be rated at 0.55 as a sound absorber at this pitch. This means that a given intensity of sound of 512 vibrations may be absorbed 0.55 as effectively by a given area of this material as compared to an equal area of open window; the absorption of an open window being taken as unity.

ERNEST SZEKELY (WRITTEN). There is no question about the fact that the ventilating practice requires considerable attention from the viewpoint of noise because, as the paper points out, noise and the cost for power are the two limiting factors which have kept ventilation systems within rather narrow limits and sometimes the restriction was economically unsound.

The mechanical noises which are created by the fan and motor of course are easily treated by sound absorbing foundations and a great deal has been done recently by the manufacturers themselves to eliminate vibrations of an objectionable nature.

The subject which is of most practical interest at present is the air-borne or air-created noise which is carried on through the duct system by means of telegraphing and reverberation.

We hope that Professor Larson and Mr. Norris will continue their experiments to a final conclusion by determining the relationship that exists between pitch, amplitude and absorbing surface somewhat along the lines that Professor Sabin's experiments were carried out in connection with acoustical treatment of spaces.

The authors should be commended for the excellent way in which they treated the subject and analyzed the problem.

A. C. WILLARD AND A. P. KRATZ (WRITTEN): This is a most interesting paper, dealing with a subject of much importance to the heating and ventilating engineer. We sincerely hope that Professor Larson and Mr. Norris continue their studies in this field since the results already obtained suggest many interesting questions not answered directly in the paper. For example:

(1) Are these results applicable to larger ducts so that the data given for a 10- x 10-in. duct may be applied to such larger ducts, at the same average velocities?

(2) What kind of *noise* (there are three noise sources named in the paper) is referred to in the second paragraph of the conclusion? Does the lining reduce one *kind of noise* better than other kinds?

(3) Why use 32 holes per inch of 0.075 in. diameter, and why use lining 1-in. in thickness?

(4) Since almost completely (1-0.14) covering the acoustical lining with sheet metal did not at all impair the acoustical value of the lining, why not use a sheet of metal with fewer or even no punchings, or perhaps merely apply the acoustical layer to the outside of the duct with a few holes drilled in the duct walls?

(5) Why not make the comparisons of lines and unlined ducts on an economic basis, with both ducts of such an area as to give the same acoustic or sound level of performance? In other words, how large an unlined duct should be constructed to give the same acoustic results as a given size of lined duct, and what would be the relative cost? Thus, the upper curve in Fig. 8 of the paper indicates that an unlined duct gives satisfactory acoustic performance at 41 decibles (auditorium conditions), or at 1,750 fpm air velocity. Hence, by merely increasing the 10- x 10-in. duct to 12- x 12-in. as required to install the lining, the air velocity, for a given air quantity, becomes 70 per cent of the velocity in the smaller duct. This would reduce a *noisy* velocity of, say 2,520 fpm to 1750 fpm (satisfactory for auditorium conditions) and save the cost of the lining and its installation, to say nothing of the power cost of operating the fan.

(6) Would it not be proper and more equitable to simply use the larger 12- x 12-in. duct without any lining, as the base for all comparisons between the unlined and the lined 10- x 10-in. duct, since the extra 2-in. space must be provided?

(7) What effect would elbows cause in sound transmission through ducts, and would the acoustic insulation of the elbows alone be more or less effective than the insulation of the straight duct?

In conclusion, it seems to us that one of the important results of this work is the upper curve in Fig. 8, which is of much interest and value to the duct designer, since it shows a *safe* noise level for auditorium conditions of about 1,750 fpm air velocity in *straight unlined sheet metal ducts.*

C. M. ASHLEY: The authors of this paper have done some very interesting experimental work and the results of their investigation should be of decided interest to members of the heating and ventilating profession. It appears, however, that the results of their studies do not lend themselves well to generalization. While their work unquestionably indicates the conditions existing in the test set-up and with the method of test which they have outlined, it is evident that it would be difficult to draw many general conclusions concerning the performance of other ventilating systems. The reasons for this are given in succeeding paragraphs.

In considering the measuring equipment, the writer finds no mention as to whether a filter network, such as that mentioned in the paper of Mr. Stanton, was used to properly simulate the frequency-sensitivity of the car. Unless such a network was used the condenser microphone will give too much weight to sounds of low frequency and very high frequency. Another point concerning the measuring apparatus of considerable importance is the location of the microphone. This, we note, has been placed several feet to one side of the outlet. The result of this is that the sound intensity measured is not actual intensity at the outlet but is a lower intensity, due to the dispersing of sound. It is well known that the sound leaving a radiating source such as the end of the duct has considerable directional effect, the higher frequency sounds being more directional than the lower frequency. For this reason the tendency of the location chosen is to under-estimate the effect of the sound absorptive material, since the sound absorptive material is more effective for the high frequency than for the low frequency sounds. There is a further effect which may or may not be of importance, due to reverberation in the room which will act to increase the sound intensity at the microphone. Since the reverberation is not uniform for all frequencies of sound, the effect of the room on the microphone will be different for different frequencies. Our experience would seem to indicate that it is possible in large ducts to obtain audiometric readings within the duct and that in both audiometric and instrumental observation the errors are kept to a minimum for small ducts by measuring the sound directly along side of the air-stream or through a diaphragm which is located in the duct.

Possibly the most serious difficulty in attempting to generalize from this test is the

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apparent lack of distinction between fan and duct noises between which there has apparently been no attempt to differentiate. The values shown for the all unlined duct unquestionably involve the fan noise as by far the most important, however, in the lower readings obtained with duct linings the duct noise probably plays some part so that here the noise is a composite of that coming through from the fan and that being generated in the duct. We believe that the separation of these elements is vital to a correct evaluation of either. Considering the results with the different linings we are inclined to question the general applicability of the conclusion which would be reached from these tests that a lining is more effective near the fan than near the outlet. This may be true where, as in this case, the outlet is directly from the end of a straight duct. In actual practice, however, this is seldom the case and we believe that the modifying influence of elbows, pan outlets, splitters, etc., may change this conclusion quite radically. There are a number of factors such as the effect of the open duct end as a sound absorber and the reverberation in the duct and the reflection from the elbows and fan which are important in determining the effectiveness of sound absorptive material and unless these are all taken into account, it is difficult to draw conclusions of wide applicability. Another factor which may have effect is the possibility of increased duct noise generation at the end of the duct, due to the presence of sound absorptive material, particularly to the perforated metal covering, although the magnitude of the differences between the location of the sound absorptive material at the beginning and end of the duct seems to preclude the possibility of this being the sole difference.

Discussing the problem which the authors are attacking in a more general way, it appears that continued experiments and observations will make possible the prediction of the sound level to be obtained in a ventilated or air conditioned space with a given duct layout, fan resistance and type of fan and sprays. This will involve the setting up of certain standard requirements for the noise generated by the fan and sprays which eventually should be written in the specifications. Applied to this will then be certain factors for the noise attenuation in the duct system and noise attenuation from the outlets to the room. From this it will be indicated what further steps are necessary to produce a noice level which is recognized as satisfactory for the particular type of room being conditioned. This method of estimating sound has already been put upon a semi-rational basis and further research coupled with field tests will undoubtedly make possible the prediction of sound intensities just as accurately as we now predict the heat load to the conditioned space. For this reason any research, such as that embodied in the present paper, is of very great value, but at the same time it must be subjected to searching criticism in order that misleading conclusions be avoided.

H. P. REID: In this paper apparently no attention has been given in the investigations to the size or to the shape of the duct. These ducts as illustrated have all been rectangular or square in cross section. Therefore, we would expect the greatest diaphragm effect of the duct walls. What would be the effect if these investigations were carried on with a round section duct instead of a rectangular or square duct? Would the diaphragm action of the duct wall be deadened to a large extent and consequently the amplification of the various types of noise media, whether it be from air vibrations or from mechanical noises set up, be affected thereby?

The sources of most of these noises have been taken either from the mechanical features of the fan or from the mechanical and electrical features, or from the vibrations in the duct. How about in the multi-duct system, as the multi-outlet system, where we have outlets in several different rooms from the same ducts or the same ventilating system? I have in mind one case in a high school where a number of teachers were very much perplexed to have the principal tell them just what happened in their rooms and just at what hour certain things were said in the room. Checking

on the principal they found he was somewhere else entirely in the other part of the building. This occurred frequently until one day one of the teachers happened to be called to the principal's office and while waiting for the principal heard what was going on in the room on the second floor below, as the noise was being conducted through the duct from openings in the same duct.

Going back to the first paper, it probably would apply equally to the paper this afternoon, one source of mechanical noise that is found in unit ventilators and frequently in the larger fan ventilating systems is the vibration on the shaft. A new sleeve bearing when properly oiled, will run smoothly and probably fairly noiselessly but if the same bearing were made with some anti-friction metal and were tightly adjusted so that the vibration be stopped, it might run fairly well without lubrication. On the other hand, most of you probably have witnessed the case of high noise in a system with a fast-rotating wheel, where the use of a few drops of oil on the bearing deadened the noise almost entirely. A small clearance of the sleeve over the shaft will allow considerable vibration on the shaft. Has any attention been given to the effects of different types of lubrication and the different types of bearings on the noise transmission through the system?

One speaker gave the analogy of a new automobile engine as against an old one—the old duct system and the old ventilating system. I say *old* after it has been in service for a few weeks. Vibration is set up by dust and grease accumulations on the fan blades. This is especially true with the propeller type wheel. Perhaps in some of these systems the noise troubles may be investigated from this standpoint.

J. D. EDWARDS: There is one factor which has not been given consideration yet which perhaps may play a part in vibration in ventilating ducts and that is the kind of metal used in their construction. To illustrate I will describe briefly an experience we had in equipping the new Aluminum Research Laboratories in New Kensington. It was the intention to have the chemical benches all made of aluminum and with one exception they were so made. The extra desk was made of steel. They were all painted with aluminum paint, however, so you could tell no difference between them. We found, however, when they were put in use that there was a very obvious difference in the noise made by pulling the drawers in and out. The steel equipment made a great deal more noise and the difference was not discovered by us but by a manufacturer of steel equipment who came to the laboratory to inspect these desks.

It may be that the lower density and lower modulus of the aluminum was more or less effective in eliminating the vibrations. We have no good explanation but it seems to me that this is one factor which may play a minor part in these various phenomena that would be well worth looking into.

E. B. LANGENBERG: In studying the problem of noise in residence work, we have found a number of conditions that have been rather difficult to solve. One of these is in the construction of square ducts.

Noise starting at the motor or fan apparently caroms at certain angles in duct work and if the ducts are not properly braced or sound-proofed, the vibration may be amplified.

We have found that should such a vibration be in tune with a string in a piano, that the string will vibrate and add to the tone already produced. A heavy picture suspended by wire would produce the same effect.

To off-set this, we have found it necessary to break the angles of the duct work, change the velocities and in several cases, increase the size of the ducts. The latter being done to dissipate the pulsations of air as they leave the fan and diffuse them to such a point that they do not set-up any vibration in the metal ducts.

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The type of fan used has much to do with the noise developed. Any fan that creates heavy pulsation of air on its exhaust side is not suitable for residential work. The air flow must be continuous and smooth.

In the duct work all joints must be perfectly smooth to prevent any turbulence in the pipe as that may cause vibration. Canvas connections must be straight and the connection to the metal work should be smooth. Duct supports should be insulated, otherwise they may carry the sound into the frame of the building and thus transmit the noise to remote locations.

Our experience has been that all duct work should be insulated for at least 12 ft on either side of the fan and that all bracing should be rigid and practically vibrationless.

R. F. NORRIS: Realizing that we are short of time I shall answer one or two of Professor Kratz' questions briefly.

His first question was, why not increase the size of the unlined duct to 12 x 12 in., thus increasing its area 44 per cent and reducing the air velocity for the same delivery.

This is a valid criticism when the ducts are small in area, say 10 x 10 or 12 x 12 in. If the ducts are large an increase of 2 in. on both dimensions will only increase the area slightly, and the consequent reduction of velocity will be negligible from an acoustical standpoint, while in the large duct the acoustical material will work almost as well as it does in the smaller ducts. In large ducts the reduction per foot of length is less than in small ducts. We have no formula by which this may be expressed, but apparently the reduction is inversely proportional to the hydraulic radius of the duct; that is, it is proportional to the ratio of the exposed surface to the cross sectional area of the duct. A larger duct must have proportionately longer lengths of acoustical lining to get the same quieting effect that would be obtained in a small duct.

The noise meant in the second paragraph was the total noise. No differentiation was made between the eddy current in the air and the fan noise. It consisted of all the noise coming through the duct. We know that acoustical materials are more efficient at some particular frequencies than at others, but disregarding that, all of the noise was cut down.

Professor Kratz also wanted to know why we use 32 holes per square inch, why they were 1/16-in. holes and why we used a 1-in. layer of acoustical material. These materials were used because they were available, the 1/16-in. holes because (although we would have liked to use smaller holes) the punches became so fragile that the expense of the material went up out of all proportion because of punch breakage. In the experimental work it was found that we could reduce the number of holes to about 1 hole for every 1¼ sq. in. without getting into serious difficulties. As soon as we made the holes so far apart that the distance between holes is the same order of magnitude as a quarter wave length of the sound, we begin to get reflection of that sound. Consequently, holes spaced 1 in. apart would give no serious difficulty.

I think it is rather obvious why we did not use plain metal with no holes. The holes in that case would be much farther apart than a quarter wave length. If the blanketing material, is put around the duct and holes are punched through the metal, I do not see that the result will be different from that which we have obtained. In a 10- x 10-in. duct the other procedure was more convenient.

There should be no more noise set up in a properly designed elbow than there is in a similar length of straight duct. In treating elbows we find best results are obtained by putting a certain amount of acoustical material right after the elbow on the discharge side to absorb the noise as quickly as it is generated.

No. 885

AIR CONDITIONING AS APPLIED TO FURNITURE, FIXTURES AND OTHER INTERIOR WOODWORK

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MEMBER

IN ITS application to human comfort and health, the art of air conditioning has, in recent years, been developed to a high degree of refinement. Scarcely less important has been the growing appreciation on the part of ventilating engineers of the importance of air conditioning in connection with industrial processing. One of the most valuable applications of air conditioning knowledge has been in connection with the fabrication and use of the thousands of articles made from wood. The purpose of this paper is to point out some of the considerations involved in the control of atmospheric temperatures and humidities in their relation to the manufacture and utility value of furniture, fixtures and other interior woodwork.

PURPOSE OF AIR CONDITIONING

Wood is a hygroscopic material composed largely of cellulose. In common with other cellulose products, wood has the property of absorbing moisture from the air or giving up moisture to the air until a condition of balance is attained between the moisture content of the wood and the relative humidity of the air. Within the range of ordinary atmospheric temperatures, this condition of balance is well below the point at which the wood fibers are saturated and the fibrous structure of the wood has the property of swelling as it absorbs moisture and of shrinking as it gives up moisture. The practical function of air conditioning as it relates to woodwork is therefore to surround the wood with such a uniform atmospheric condition that this shrinking or swelling will be held to a negligible amount and the wooden article will be maintained at an approximately constant size.

In addition to changes in size, swelling or shrinking of the wood in response

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to changes in atmospheric humidity may also set up fiber stresses in the wood. If the moisture changes are fairly uniform, these stresses are negligible. When a wide variation in moisture content develops in different parts of the piece of wood, however, as between the two faces or between the ends and the middle, more or less distortion of shape may occur, such as cupping or twisting.

Fortunately, the atmospheric changes within buildings are ordinarily so gradual from season to season that distortion in furniture or woodwork due to fiber stresses seldom occurs under the usual conditions of use. In extreme cases, where the wood is subjected to exceptionally moist conditions, followed within a short time by exceptionally dry conditions, severe stresses may develop amounting to several hundred pounds per square inch. Separation of joints and warping are not surprising under such circumstances and may be sufficient to damage the appearance or utility value of the article concerned.

In the design and fabrication of wooden articles and woodwork, the tendency of the wood to shrink and swell with moisture changes is taken into consideration and provision is made to permit a certain amount of shrinking and swelling without injury. These provisions are not sufficient, however, to accommodate all of the changes in size and shape that may occur in the seasonal swing from outdoor conditions in the summer time to indoor conditions in heated buildings during extremely cold weather such as prevails in the northern and central portions of the United States. The assistance of the air conditioning engineer is therefore helpful to prevent damage to the furniture and other woodwork due to excessively low humidities in heated buildings as well as to prevent the development of conditions harmful to the mucous membranes and respiratory tracts of the human occupants.

AMOUNT OF SHRINKAGE OF WOOD

The shrinkage of wood shows a considerable variation in accordance with the species and density and the direction of the fibers. In changing from a condition of fiber saturation (for most woods approximately 30 per cent moisture content in terms of the oven-dry weight) to an oven-dry condition, with practically no moisture left in the wood, the shrinkage lengthwise of the fibers normally amounts to from 0.10 to 0.35 per cent and is negligible. Under the same conditions, the shrinkage across the grain in a radial direction (from the bark to the pith of the tree) ranges from 2.0 to 8.5 per cent of the original size, and the shrinkage across the grain in a tangential direction (parallel to the bark) ranges from 4.0 to 14.0 per cent.

On an average, the shrinkage in a tangential direction across the fibers is about twice as much as in a radial direction, although certain woods show approximately the same shrinkage in both directions. From the fiber saturation point to an oven-dry condition, the shrinkage is in direct proportion to the loss in moisture content and shows a straight line relationship. Thus it may be assumed that, below the fiber saturation point, each 1 per cent change in moisture content will be accompanied by approximately 1/30 of the total possible shrinkage or swelling. Obviously the seasonal changes to which wood is ordinarily subject will involve only a small per cent of the total possible shrinkage or swelling.

Research conducted by the *U. S. Forest Products Laboratory* in co-operation

with the *National Lumber Manufacturers Association* during the past two years has attempted to determine the seasonal changes in moisture content and shrinkage of unfinished samples of raw wood under interior conditions in representative cities in different parts of the country. The results are summarized in Table 1.

The normal shrinkage of hard maple and other cabinet woods used for

TABLE 1. MAXIMUM AND MINIMUM AVERAGE MOISTURE CONTENT OF RAW WOOD SAMPLES UNDER INTERIOR CONDITIONS, 1928 TO 1930 [a]

CITY	AVERAGE MOISTURE CONTENT (PER CENT OF OVEN-DRY WEIGHT)		
	MAXIMUM	MINIMUM	RANGE
Washington D. C.	11.0	7.0	4.0
Albuquerque, N. M.	8.0	5.0	3.0
San Francisco, Cal.	10.0	8.5	1.5
Portland, Ore.	8.5	7.0	1.5
Madison, Wis.	10.5	6.0	4.5
New Orleans, La.	13.0	11.0	2.0

[a] Matched samples of hard maple were used in this series of tests as representative of most of the hardwoods used for cabinet work.

furniture and interior woodwork, from the fiber saturation point to an oven-dry condition, ranges from a minimum of about 4.0 per cent radially to a maximum of about 11.0 per cent tangentially. Under these moisture changes, therefore, the shrinkage of raw wood would vary from 5 per cent $\left(\frac{1.5}{30}\right)$ of 4 per cent to 15 per cent $\left(\frac{4.5}{30}\right)$ of 11 per cent or from 0.20 per cent radially to 1.65 per cent tangentially. With these changes, therefore, a 12-in. board would change in width between summer and winter from 0.024 in. in a radial direction in San Francisco to 0.198 in. in a tangential direction in Madison.

Shrinkage of 0.024 in. or about 1/40 of an inch in 12 in. is negligible, of course, but a shrinkage of 0.198 or about 1/6 of an inch, in 12 in. might do considerable harm to furniture, flooring and other forms of woodwork. In addition, it should be borne in mind that woodwork is frequently subjected to greater extremes (of dryness in the winter and high humidity in the summer) than the average conditions surrounding the samples in the tests summarized in Table 1. Buildings often become overheated during the winter season. As a result, humidities may drop low enough to produce a moisture content as low as 4.0 per cent in woodwork. At the other extreme, an unusually long period of hot, humid days in the summer time may be sufficient to build up the moisture in the wood to as much as 15 or 16 per cent for temporary periods.

Thus the shrinkage and swelling of the wood may, under some circumstances, be more than twice as great as indicated by the figures in Table 1. While these severe conditions are more or less unusual and abnormal, they are none the less harmful when encountered and should be considered. It is in such cases particularly that air conditioning, by diminishing extreme seasonal variations in atmospheric humidity, can be made a factor of real importance.

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PROTECTION OF FURNITURE AND WOODWORK

One valuable means of protecting furniture and woodwork from damage due to moisture changes is the use of moisture resistant finishes, such as varnish, lacquer, enamel, and paint. While these finishes are all more or less porous and subject to moisture penetration, they retard the rate of moisture changes in the wood sufficiently to materially reduce the extent of the seasonal changes between summer and winter. Thus varnished or painted furniture or woodwork will show decidedly less shrinkage and swelling than unpainted raw wood exposed to the same conditions.

Moisture in finished woodwork will show a *lag* of several weeks in its response to atmospheric moisture changes and this lag may be sufficient to reduce the total seasonal variation to less than half that of unfinished wood. Much depends upon the thoroughness of the finishing and the extent to which a protective coating is used on the back side or inner surfaces of the wood. One coat of varnish on the outside surface with no back-painting or priming is of minimum value. Two coats are considerably more than twice as effective as one coat, since the second coat serves to cover the minute voids left in the first coat. Each additional coat, up to 12 coats, has been found by the *U. S. Forest Products Laboratory* to give added moisture resistance (Technical Note No. F-12). After the fourth coat, however, the added value of the further coats was shown to be comparatively small.

Perhaps the most conspicuous examples of the value of atmospheric humidity control in the protection of furniture and woodwork occur in connection with antique furniture, wall paneling, and other woodwork imported from foreign countries. American museums and wealthy individuals have imported a considerable quantity of such material in recent years, including chairs, benches, chests, mantels, wall panels, doorways, stair railings, plank floors, beamed ceilings, desks, tables, and other articles made of wood. Often whole rooms have been removed from English and French mansions to be reassembled in American museums, homes, and private offices. Considerable damage and disappointment have resulted from the unexpected shrinkage of this furniture and woodwork when subjected to the low humidities commonly found in artificially heated American buildings in the colder sections of this country.

Atmospheric humidities, both out-of-door and indoor, are much higher in Western Europe than in most parts of the United States and the less efficient methods of heating generally employed result in considerably lower indoor temperatures during cold weather than are customary in our homes and offices and public buildings. Consequently, old furniture and woodwork comes to a balance with atmospheric humidities in European buildings at moisture contents that are rarely under 9 per cent and may be as high as 14 or 15 per cent, based on oven-dry weight, during certain seasons of the year when rain and fog are particularly prevalent.

Woodwork in this condition cannot be subjected to the low atmospheric humidities characteristic of the winter time without an appreciable amount of shrinkage, which is often sufficient to cause damage. Humidities as low as 25 or even 20 per cent are not uncommon in steam-heated buildings in this country in sub-zero weather, where no provisions have been made for humidity control. Such low humidities may dry woodwork and furniture to a moisture

content below 5 per cent if continued for several weeks. Under such circumstances, wooden articles brought over from England and France might lose as much as 8 or 9 per cent in moisture content with a shrinkage as high as 2 or 3 per cent of their original dimensions.

DAMAGE TO VALUABLE FURNITURE AND WOODWORK

Among the numerous cases of injury to imported furniture and woodwork from shrinkage that have occurred in this country, the following might be cited as typical. The directors' room of a large western railroad company in a Chicago office building had been furnished with a large imported Chinese table made of teakwood inlaid with ivory and elaborately carved. The outer edge of the table top was constructed of four solid wood members, mitered at the corners, the inner portion of the top consisting of other pieces of solid wood separated from the outer framework by an inland ivory design. After this valuable table had been subjected to one winter of dry, steam-heated air, the wood members of the top shrunk sufficiently to cause the mitered joints at the corners to open up in a V-shaped crack and to cause a noticeable crack between the outer framework and the interior portion of the table, causing the inlaid work to separate and loosen. The inner boards of the top also developed cracks in the joints.

Another instance occurred in Washington, D. C. An enthusiastic collector of rare and beautiful examples of old furniture had gradually accumulated a large collection of pieces picked up in European mansions and castles. These were kept stored in a Washington warehouse and remained in good condition while in storage because the warehouse temperatures during the heating season were considerably lower than living room temperatures and humidities correspondingly higher. Some two years ago, a residence was completed and furnished with these treasures.

One can imagine the owner's dismay during his first winter in the new home when he found that the dry winter atmosphere was drying his prize pieces to the point where joints were opening up, cupboard doors and drawers were getting loose, panels were shrinking enough to expose interior wood around the edges, and certain parts were showing a tendency to twist and warp. Realizing the cause of the trouble, this owner promptly installed a small air conditioning plant and has eliminated the damage to his furniture by maintaining winter humidities of at least 40 per cent. Incidentally, he has found that these higher humidities make the house comfortable at lower temperatures and also prevent the shrinkage of floors and woodwork as well as furniture.

PREPARATION OF LUMBER

Manufacturers of furniture and interior trim in this country have learned to avoid much of this shrinkage trouble by kiln-drying their lumber to a low moisture content—usually from 4 to 6 per cent of the dry weight—before fabrication. The best practice is to store this dry lumber in heated storerooms until it is used, in order to prevent it from re-absorbing moisture from the air, since average humidities in unheated storage will cause a regain up to from 10 to 14 per cent over a period of several months. Articles of furniture and woodwork made from lumber in such a condition are very apt to develop shrinkage troubles when used in heated buildings in the colder sections

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of the United States. Manufacturers having wide-spread distribution try, as a rule, to use lumber that is dry enough to meet the needs of heated interiors in cold climates.

In this attempt to avoid shrinkage troubles, however, it is quite possible to use lumber that is so dry that it will give difficulty from swelling. Sticking drawer fronts, bulging and sticking doors, warped tops, and buckled floors are some of the annoying results of swelling caused by an excessive absorption of moisture. It is true that proper design will accommodate practically all of the swelling that is apt to occur under normal conditions of use, even in the warm, humid summer atmosphere. Unusually high humidities are apt to occur occasionally, however, which may cause trouble from swelling, and it is just as well to avoid the risk of such damage by avoiding the use of over-dry lumber.

The figures in Table 1 will serve as a reliable guide to the proper range in moisture content of lumber used for furniture and other interior woodwork. Obviously, the lumber should have a moisture content more or less mid-way between the winter minimum and the summer maximum. Since damage from shrinkage is apt to be somewhat more serious than that from swelling, the optimum condition would be below rather than above the mean between the two extremes. Taking the figures for Madison, Wis., as typical of the colder sections, with a variation from 6.0 per cent to 10.5 per cent, it would appear that lumber containing from 7 to 8 per cent would give the least trouble from shrinkage and swelling under average conditions of use.

In practice, it is advantageous to dry the lumber to a cross-sectional average of from 5 to 6 per cent in which the surface might be as low as 4 per cent and the extreme center as high as 7 per cent. Surface regain will then occur during the normal period of storage and fabrication, except in extremely cold weather, sufficient to bring the cross-sectional average to between 7 and 8 per cent and uniform throughout. Control of humidity in factory and store-rooms is needed only to the extent of preventing the stock from regaining much above 8 per cent.

EQUILIBRIUM MOISTURE OF NATIVE WOODS

The balance between atmospheric moisture and moisture in wood for any given temperature and humidity has been shown by the *U. S. Forest Products Laboratory* to be practically the same for the four species of native cabinet woods included in the test. The equilibrium moisture content of these four species at a dry-bulb temperature of approximately 70 F and relative humidities from 20 to 95 per cent are shown in Table 2, the figures and data being taken from Technical Note No. F-15 published by the *Forest Products Laboratory.*

TABLE 2. EQUILIBRIUM MOISTURE CONTENT OF FOUR NATIVE HARDWOODS

Species of Wood	Per Cent Relative Humidity (at 70 F)					
	20	35	50	65	80	95
White Oak	4.5	7.0	9.5	12.6	17.5	26.5
Black Walnut	4.8	7.0	9.4	12.3	16.7	24.4
Yellow Birch	4.2	6.5	9.2	12.4	17.0	26.0
Ash	4.3	6.7	9.7	13.2	18.0	25.2

Based on research conducted with several commercial species of softwoods and hardwoods, all of which have been found to respond in approximately the same degree to atmospheric moisture changes, the *Forest Products Laboratory* has prepared a chart, still subject to minor revision and correction, showing tentatively the equilibrium moisture content of wood over a wide range of air temperatures and humidities. This chart was published in 1928 by the *National Committee on Wood Utilization* of the U. S. Department of Commerce, in the Distributor's Edition of their bulletin entitled The Seasoning, Handling, and Care of Lumber. As a guide to engineers interested in the effect of atmospheric conditions on the moisture in wood, a portion of the data contained in this chart has been reduced by the writer to tabular form for temperatures from 50 to 120 F and for wet-bulb depressions from 4 to 30 deg as shown in Table 3.

TABLE 3. EQUILIBRIUM MOISTURE CONTENT OF WOOD FOR VARIOUS AIR TEMPERATURES AND WET-BULB DEPRESSIONS APT TO BE ENCOUNTERED IN AIR-CONDITIONING WORK (FROM DATA COLLECTED BY *Forest Products Laboratory*)

Wet-Bulb Depression (Deg F)	Per Cent Moisture Content (in Terms of Oven-Dry Weight) Dry-Bulb Temperature (Deg F)							
	50	60	70	80	90	100	110	120
4	15.8	16.7	17.4	17.9	18.2	18.3	18.4	18.5
5	14.0	14.9	15.7	16.2	16.5	16.7	16.8	16.9
6	12.5	13.5	14.3	14.8	15.2	15.4	15.5	15.6
7	11.2	12.3	13.2	13.7	14.1	14.3	14.4	14.5
8	10.1	11.3	12.2	12.8	13.2	13.4	13.5	13.6
9	9.1	10.3	11.3	11.9	12.4	12.6	12.8	12.9
10	8.1	9.4	10.4	11.1	11.6	11.9	12.1	12.3
11	7.2	8.6	9.6	10.3	10.9	11.2	11.5	11.7
12	6.4	7.8	8.8	9.6	10.2	10.6	10.9	11.1
13	5.6	7.1	8.1	8.9	9.6	10.0	10.3	10.6
14	4.8	6.4	7.4	8.3	9.0	9.4	9.8	10.1
15		5.8	6.8	7.7	8.5	8.9	9.3	9.6
16		5.2	6.2	7.2	8.0	8.4	8.9	9.2
17		4.7	5.7	6.7	7.5	8.0	8.5	8.8
18			5.2	6.3	7.1	7.6	8.1	8.4
19			4.8	5.9	6.7	7.2	7.7	8.0
20				5.5	6.3	6.8	7.3	7.6
21				5.1	5.9	6.5	7.0	7.3
22				4.7	5.6	6.2	6.7	7.0
23					5.3	5.9	6.4	6.7
24					5.0	5.6	6.1	6.4
25					4.7	5.3	5.8	6.1
26						5.0	5.5	5.8
27						4.7	5.2	5.5
28							4.9	5.2
29							4.7	5.0
30								4.8

The figures in Tables 1, 2, and 3 represent averages. Individual pieces of wood will vary as much as 2 per cent from these averages, due to the characteristics of the particular piece of wood concerned and slight variations in the atmospheric conditions to which each piece is exposed. For example, wood close to a window is apt to have a higher moisture content than that close to an interior wall; also samples exposed to direct radiation of a radiator or heating element will be somewhat drier than wood not so exposed.

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The data summarized in Table 1 are of interest from a heating standpoint as the average maximum and minimum figures are the composite result of climatic conditions of the city concerned as modified by the methods of heating used in that locality. For example, it was found that in the mild climate of Portland, Oregon, heat was maintained for a portion of each day for nearly 10 months of the year; yet the total hours of heat were only about 4,400. At Madison, heat is used nearly twice as many hours in the season, yet the heating season lasts only about 8 months. The comparatively low average moisture content and small variation between seasons at Portland can be accounted for by the fact that some artificial heat is needed almost every day during the cool evenings with the exception of about two months in midsummer when the atmospheric humidity is naturally low. In Portland, therefore, there is much less probability that furniture and woodwork in heated buildings will swell or shrink enough to do harm than in a climate such as Madison, with its lack of artificial heat and high humidity in the summer time and its continuous heat combined with the low dew-point characteristic of cold outdoor air in the winter time.

Another difficulty encountered in the colder climates of the central and eastern parts of the United States is the condensation that is liable to occur if an attempt is made to maintain a humidity sufficiently high to prevent the harmful shrinkage of furniture and woodwork. If the interior air contains very much moisture, a low outside temperature will cause excessive condensation on the inside of the windows unless these are protected by tight storm-sash. Condensation is apt to occur on walls of average heat transmission if humidities much above 40 per cent are present when the outside temperature is zero or below.

It is evident, then, that the maximum humidity which it is practicable to maintain in heated buildings in cold weather is directly dependent upon the out-door temperature. The lower the out-door temperature, the lower must be the interior humidity, even if means are available to humidify the interior air. In the case of museums, housed in monumental buildings with very thick walls, this will not be as true as in ordinary dwelling houses, but even in museums care must be exercised to prevent the humidity from reaching a point where condensation will occur. Fortunately extremely cold weather is usually of short duration and the use of a higher humidity between cold spells will usually be sufficient to prevent any harmful shrinkage in furniture and woodwork that is protected against rapid moisture changes by paint or varnish.

Even in the coldest weather it is usually possible to maintain a relative humidity of at least 35 per cent. At 70 F, this will ultimately dry the wood to a moisture content of 7 per cent if continued for several weeks. The removal of the last 1 per cent is so slow, however, in varnished or painted furniture or woodwork, that such articles are not apt to dry much below 8 per cent during the duration of the usual period of cold weather. By raising the humidity of at least 35 per cent. At 70 F, this will ultimately dry the wood effect of the dryer periods is more or less compensated by intermediate periods of re-absorption which will tend to prevent the wood in a humidified building from drying below 8 per cent even in a cold climate. This would not be true, however, in buildings not provided with adequate arrangements for adding moisture to the interior air as needed.

The problem of preventing excessive absorption and swelling in the summer is not so simple, as this is only possible where facilities are available for dehumidification of the interior air. Where such facilities are available, as in museums, department stores, and office buildings, a relative humidity averaging not higher than 60 per cent and preferably about 50 per cent will keep woodwork in excellent condition and confine swelling to a negligible amount. Fortunately, as pointed out before, a normal amount of swelling is provided for in the design of most furniture and woodwork so that the probability of damage during exposure to summer humidities is slight, particularly if these articles are not allowed to become excessively dry during the winter heating season.

DISCUSSION

W. KARL LAUGHBOROUGH (WRITTEN): An appreciation of the importance of the effects which air conditions have on the behavior of wood used in buildings is shown by the numerous requests we receive for this type of information. Mr. Brewster's article seems timely and indicates that he not only recognizes the importance of his subject, but also that he is in possession of a fund of knowledge that should be valuable to those interested in the utilization of wood.

Our data show that interior finish, flooring, and the like are often so severely stressed perpendicular to the grain by a steep moisture gradient (higher moisture content on the surface than in the interior) that the elastic limit in compression perpendicular is exceeded. On redrying boards so affected, objectionable cracks appear between them. The phenomenon depends upon the intensity of the stress and upon the elastic limit of the species involved perhaps more than it does on the rate of moisture regain. A preliminary analysis of our data indicate that when each piece of flooring is uniformly dry in cross section and snugly laid, a regain of 4 per cent moisture content, uniformly distributed, will cause cracks, when the floor redries to the moisture content it had when laid. Such moisture changes in flooring might easily occur as the result of the normal seasonal cycle of atmospheric conditions in a northern residence.

When dry flooring is closely laid in a church basement for example, where the relative humidity is apt to be high, the floor may actually bulge. Of course, it may not be expedient to control the relative humidity in churches throughout the week, but obviously some notion of the average relative humidity within them should be known in order that the lumber may be appropriately dried for the use. The author indicates that the lumber should be dried and assembled at a moisture content in keeping with the average relative humidity to which the wood is exposed. While 7 per cent moisture content may be a satisfactory value for most interior uses, certain use requirements demand a higher moisture content. We are of the opinion that the design and fabrication of wooden articles should take into account the tendency for dry wood to swell when it is to be exposed to relative humidities as high as those normally found in churches and halls that are heated only periodically during the winter. Perhaps the lumber used in such places should be assembled at a moisture content of 10 instead of 7 per cent.

One important phase of wood-humidity relationship in buildings, which probably falls outside the scope of Mr. Brewster's paper, involves the difficulties encountered in the use of wood in steel and concrete construction. The haste of finishing a building of this kind and getting it occupied often demands that dry lumber be assembled in the floor and walls of rooms which are made humid by fresh plaster and concrete. The heating and ventilating engineers may be of service in solving the problem.

Serious condensation problems arise because the dewpoints of adjoining rooms are not properly controlled. The backing of many floors rot when a highly heated room is situated over a relatively cold room. This type of problem logically falls within the province of the heating and ventilating engineers. It involves an understanding of temperatures in partitions, floors, or ceilings, as the case may be, and the relation of the relative humidity at these places and the moisture content of adjoining wood. As the author has stated, the dewpoint of the walls largely determines the relative humidity that may be maintained within. If the temperature of the walls in a frame building is below the dewpoint temperature of the rooms within the building, dangerous condensation problems arise that tend to encourage decay. Heating and ventilating engineers are in a position to so govern the temperatures and relative humidities in different rooms that the temperatures of the walls and partitions are not below the dewpoint. Further advance along this line demands the joint effort of the architects and the heating and ventilating engineers. The one to provide the proper insulation in the walls, ceiling, and roof, and the other to control the temperature and relative humidity in accordance with the temperature limitations set by the architect's design.

I have nothing but praise for Mr. Brewster's paper and I wish merely to emphasize the importance of controlling atmospheric conditions with the view of avoiding compression set and resultant unsightly cracks together with the importance of controlling the temperature of the walls, partitions in such a way as to prevent serious condensation problems.

J. D. EDWARDS: The author has made a timely contribution on a subject which, while not particularly obscure, nevertheless has not received the consideration which it should have in some quarters. The facts he has disclosed represent an opportunity for heating and ventilating engineers to correct the conditions cited. The lumber industry, lumber manufacturers and those who utilize wood, have been busy attempting a solution of the problem. The Forest Products Laboratory has been active in investigating all the factors which are pertinent to a change in property of wood with change in moisture content.

About 7 or 8 years ago an investigation was started to determine the moisture-proofing properties of various coatings. This work commenced during the war when it was desired to find a coating for wood airplane propellers which would absolutely moisture-proof them. This work has been continued since that time and one of the most interesting things they have discovered, at least interesting to the Aluminum Research Laboratories, was the fact that aluminum bronze powder incorporated in the usual paint vehicles greatly increased the moisture-proofing power. For example, one coat of the ordinary interior varnish has a moisture-proofing efficiency of 17 per cent. One coat of aluminum paint, which might be substituted for it, would have a moisture-proofing efficiency of 60 per cent, three coats 90 per cent.

These facts naturally have been interesting to the wood manufacturers and at the present time concrete application of these is being made. For example, the new Sterling Laboratory at Yale University has a great deal of woodwork in it and the bookcases are all backprimed with aluminum paint in order to equalize and minimize moisture changes in the wood.

Manufacturers of window sash and trim and other products going into house construction have recognized this difficulty and a number of them have already started priming this material at the mill with aluminum paint so that during the time it is erected it is properly protected against moisture changes.

Perhaps the newest development in this line is what is called the mill-priming of lumber. Steel, since time immemorial has been primed at the mill with red lead or with some other priming coat to protect it against rusting. Protecting the wood against moisture changes by priming at the mill is now a common practice. The use of moisture-proofing coatings is only a partial solution of the problem.

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No. 886

AIR CONDITIONING IN THE BAKERY

By W. L. Fleisher,[1] New York
MEMBER

THERE is no doubt that air conditioning is essential to bread-making because in some simple manner temperature and moisture have been an essential part of kneading, fermentation and baking of bread since primitive times. But the writer has been forced to change his ideas drastically as to just what air conditioning does do in a bakery, after a thorough investigation of the basic electrical and chemical laws which underlie this subject. The industrial application of air conditioning to any field revolves principally around moisture regain and static electricity. Probably there is more connection between these two factors than the air-conditoning engineer has suspected. While the conditions required for the production of a good loaf of bread are generally known, it is not entirely obvious as to why these conditions have the effect that they do. It is probable that the treatment of bread is almost entirely a surface problem, and consequently the air-conditioning equipment for bakeries which has been designed (as most of it has been) to deal with the entire bulk either of the flour, the dough or the loaf is not only unessential, but possibly deleterious to production. This probably explains the reasons why the simple methods of conditioning which were part of the bread-making art for so many centuries and which consisted principally in maintaining a suitable temperature for fermentation or the retarding of fermentation, and such simple expedients as dampened cloths, etc., which were used to cover the doughs during the fermentation period, were so efficacious, because only the exposed surfaces of the doughs and the breads were affected by exterior conditions, when the time element was taken into consideration.

Bread is essentially a combination of wheat flour and water to which has been added salt, yeast, sugar, malt extract, milk and shortening. The average batch of bread contains the following amounts of various ingredients: flour, 300 lb; water, 180 lb; salt, 5½ lb; sugar, 7 lb; malt extract, 3 lb; yeast, 6 lb; shortening, 6 lb; milk 7 lb. The percentages of each of the ingredients to the whole batch is the result of hundreds of years of experimentation to obtain satisfactory results.

It has been stated that industrial air conditioning has principally to do with

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the regain of moisture. In other words, that there is a set moisture content for every hygroscopic material which follows the line or curve of relative humidity; that with a change in the relative humidity, the hygroscopic material loses or gains moisture and that equilibrium is established when the vapor pressure of the atmosphere surrounding it equals the vapor pressure of the hygroscopic material. Air conditioning authorities have even followed this line of reasoning in connection with dough and bread, but it does not apply where the substance treated to a bath of conditioned air is in a colloidal state. Consequently the behavior of doughs and bread and even the flour, which is the main constituent in dough and bread, does not follow this reasoning, as the flour itself, to a more or less degree, is in a colloidal state from its own moisture content, which it holds and retains from the wheat berry.

HYDRATION PROCESS IN MODERN BAKING

As bread-making is to a large extent a hydration process, and as the actual moisture content is little affected by the surrounding atmosphere, there may be some question as to why air conditioning is so essential. The modern machine bakery consists of a flour storage department in which the flour, either in sacks or barrels, is stored for considerable time. It has been shown by innumerable tests that a temperature of about 78 F and a relative humidity of 70 per cent are very satisfactory for the storing of flour. Under these conditions, flour will retain about 13 per cent water by weight. However, if flour is dried for several hours at a temperature as high as 212 F with a correspondingly low relative humidity, the flour still retains 11 per cent or thereabouts of moisture by weight.

The problem of moisture in flour is further complicated by the fact that the water is united to the solid particles in varying degrees of affinity and also by the fact that there are two kinds of moisture in flour, sometimes called *moisture of constitution* and *moisture of wetness*. The moisture of wetness is the moisture which adheres to the surface of the flour particles and is purely physical. Violent heat will very quickly vaporize, disperse and drive off this water, leaving the flour itself more or less unaltered. The moisture of constitution, which is really water of hydration of the swollen protein particles, is of major effect in determining the useful properties of the flour. This moisture is held by a physio-chemical attraction, and if or when removed the properties of the proteins will be completely changed, although the chemical compositions remains unaltered.

Seemingly, therefore, air conditioning of the flour storage has to do simply with this one or two per cent of surface moisture, rather than the important chemical-affinity water in the protein. Fig. 1 shows graphically the conditions of temperature and relative humidity, conditions which over a period of years have proved most satisfactory to the bakers. The question therefore arises as to whether these conditions are not necessitated or recommended because of the very method of producing them, and whether air conditioning in the bakery should not be separated into its various subdivisions of temperature, moisture content and air movement, in order to determine whether each of these may not be adversely affecting the other, the results being a wasteful method of improvement.

In order to appreciate the hydration problem in a bakery—and it is, after

all, to preserve this hydration that air conditioning is advocated, or at least it is for this reason that it has been advocated up to this time—it is necessary to understand the chemistry of flour. It is the reaction of the various elements in their molecular state which go to make up the wheat berry and the resulting flour, which so profoundly affect the moisture retention possibilities of flour, and eventually of the dough. It is essential, in fact, in order to balance the results, to understand the chemistry of colloidal behavior and the action of electrolytes in order to fully appreciate the value of air conditioning in the bakery.

An investigation of moisture loss or gain from the flour to the finished

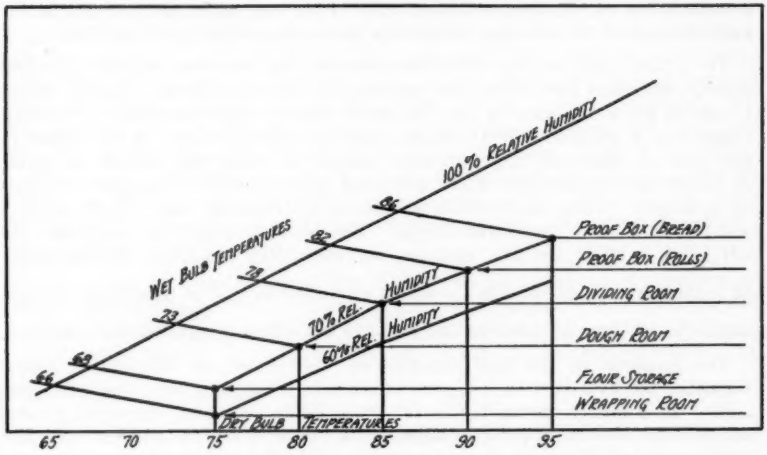

FIG. 1. PSYCHROMETRIC CHART FOR BAKERY CONDITIONS

loaf in the light of the colloidal retention of moisture would lead one to believe that the moisture change which takes place throughout the bread making process cannot possibly be anything except a surface condition.

HYDROGEN-ION CONCENTRATION

The most important basic principle in the making of bread as determined by scientists who have investigated this problem from the chemical and physical standpoint is the hydrogen-ion concentration. The writer feels that without a thorough knowledge of hydrogen-ion concentration and its effect on the starch, the gluten and the fermentation itself, a true knowledge of the effect of air conditioning on bread-making cannot be attained. An explanation of acidity and hydrogen-ion concentration should therefore prove helpful.

The acidity of a solution, which is denoted by the symbol *pH*, depends upon the concentration of hydrogen-ion derived from the dissociation of the substance in solution into its component ions. By hydrogen-ion concentration, denoted by the symbol *H*, is meant the concentration of hydrogen-ion expressed in grams equivalent weight per liter of solution. By the *pH* of a solution is

meant the hydrogen-ion concentration in grams equivalent weight expressed as the log to the base 10 of its reciprocal or, in other words, the negative log of the hydrogen-ion concentration with the negative sign omitted. Sorensen found that if the hydrogen-ion concentration of a solution is determined by means of a hydrogen electrode, the equation reduces to

$$\frac{\text{Potential}}{\text{numerical factor}} = \log\left(\frac{1}{H}\right) \qquad (1)$$

and he introduced this method of designating hydrogen-ion concentration. The pH of a solution thus designates the intensity of the acidity and not the concentration of the total available acid. One may state that the pH is the measurement of the voltage rather than of the amperage of the acidity.

To further illustrate the difference between concentration of total acid and acidity, one must first define the meaning of the term *normal solution* which is one of the terms used by the chemist to express concentrations. A normal solution is a solution which contains one equivalent weight of the solute in one liter of solutions. By equivalent weight is meant that weight in grams of an element or substance which will react with 8 grams of oxygen to 1 gram of hydrogen. Thus the formula weight of hydrochloric acid (HCl) is 36.5 and this is also its equivalent weight. The formula weight of sulphuric acid (H_2SO_4) is 98.08 and this contains two equivalent weights. The normality of a solution is expressed by $\frac{W}{E}$ where W is the weight in grams of the substance in one liter of solution and E is the equivalent weight of the substance.

The strength of the acid depends on the degree of dissociation which furnishes the hydrogen-ion. Hydrochloric acid belongs to the group of strong acids. It is highly dissociated in the more dilute solutions. In a 0.1 normal solution the HCl is 92 per cent dissociated, hence in such a solution there are 0.092 grams of H-ion per liter of solution. The pH of this solution is the log of $\frac{1}{0.092}$ or 1.04. The hydrogen-ion concentration H is written 9.2×10^{-2}. Diluting this 0.1 normal solution so as to obtain a 0.00001 normal solution a hydrogen-ion concentration H of 9.2×10^{-6} and a pH of 5.04 is obtained.

With a 0.1 normal solution of acetic acid (equivalent weight, 60) a hydrogen-ion concentration of 0.13×10^{-2} is obtained, as acetic acid is a weak acid and its dissociation in that dilution is only 1.3 per cent. The pH of a 0.1 normal acetic acid solution is 2.9, while the pH of a 0.00001 normal acetic acid solution is 6.9, practically at the neutral point. This explains why a 0.1 normal solution of acetic acid is of the acidity of ordinary vinegar, while a 0.1 normal solution of HCl would be caustic to the taste.

The point of neutrality mentioned lies at pH 7. Pure water will conduct an electric current and is therefore slightly ionized, and will furnish ions for the conductance of the current. Water (H_2O) may be written HOH and it is slightly dissociated into positive H-ions and negatively charged OH-ions. Since it furnishes the same number of H-ions which are acidic in character as OH-ions which are basic in character, it is neutral in reaction.

By the law of mass action which covers chemical equilibrium, the relation for

water is $\frac{[H^+] \times [OH^-]}{H_2O} = K$, a constant. Since the concentration of the undissociated water, even for dilute solutions up to about 0.1 normal remains fairly constant, the relation may be written $[H^+] \times [OH^-] = K_w$. The constant K_w, which is the same as K, is known as the ionization constant of water and has been determined to be 10^{-14} at 77 F, or *pH 7*. Any hydrogen concentration exceeding this will give acidity, while if the hydrogen-ion concentration decreases below this point, the solution will be alkaline. It must be remembered that as the numerical value of the symbol *pH* decreases, the acidity increases, and vice versa.

This may be explained by the law of mass action, which law states that, at a constant temperature, the product of the active masses on one side of a chemical equation, when divided by the product of the active masses of the other side of the chemical equation, is a constant, regardless of the amounts of each substance present at the beginning of the action. Therefore, the concentration of hydrogen ions times the concentration of hydroxol ions over the concentration of undissociated H_2O equals a constant. And, as the undissociated mass in the case of water is enormous, and can be taken as a constant relative to the hydrogen ions and the hydroxol ions, the equation may drop the H_2O from the equation, and have the hydrogen ion times the hydroxol ion concentration equal a constant.

Now, as in the case of water, the balance is in equilibrium, the concentration of hydrogen ions and hydroxol ions is equal, and as the product of the two equals a constant which is 10^{-14}, both the hydrogen ion and the hydroxol ion concentration is 10^{-7}. Therefore, the value of the equilibrium is the logarithm of $\frac{1}{10^{-7}}$ or *pH 7*.

S. P. L. Sorenson, probably the outstanding cereal chemist today, states that the *pH* of the dough is of the highest importance in insuring good bread from an otherwise good dough and the explanation as to why the concentration of hydrogen-ions affects the quality of the bread is explained by him from two angles. *First,* the hydrogen-ion concentration affects the enzymes of the flour and the yeast, and *second,* the physical-chemical condition of the substances in the flour, particularly in the proteins.

It has been definitely proved by authorities working on this subject that a *pH* concentration of approximately 5 is the best *pH* concentration, both for the development of enzyme action for the production of carbon dioxide and alcohol, and for the development of the gluten for the adsorption of moisture. What bearing this explanation of hydrogen-ion concentration may have on air conditioning in the bakery cannot be determined without a more thorough knowledge of the ingredients which go to make up the baked bread.

In the earlier part of this paper it was stated that flour constituted about 60 per cent of the dough entering into the making of bread. This wheat flour itself may be divided into the following proximate analysis: starch 65 to 70 per cent; protein, 9 to 14 per cent; water, 13 to 15 per cent; cellulose and fat, 1 per cent; sugar 2½ per cent; and ash, ½ per cent. From this analysis one would assume that starch was the most important component of flour, but this is not true. During the fermentation of the dough, a small part of the

starch is attacked by the enzymes and is broken down into sugar substances, which the yeast ferments.

The gelatinization of wheat starch is of importance, because in the baking a certain amount of this gelatinization takes place. Wheat starch will commence to gelatinize and swell if placed in sufficient water at 140 F. and at 149 F all the starch turns into a starch paste. It might be assumed that as the oven temperature rises to 500 F and the amount of water in the dough is approximately 40 per cent of the total weight, all of the starch in the baked loaf is gelatinized, whereas in fact, only a very few grains are gelatinized. The reason for this is that the inside of the loaf is seldom over 200 F, and the time factor is comparatively short.

It should be emphasized that the 40 per cent of water is not available as water, as the gluten absorbs a very large proportion of it. Starch is colloidal in its nature, although the colloidal properties of ungelatinized starch are not nearly as marked as those of other ingredients of flour.

PROTEINS

The proteins which compose 9 to 14 per cent of the flour are complex substances of large molecules built up of amino acids which contain both an acid group and a NH_2 group and therefore may behave as acids or as bases depending on the *pH* and owing to the size of the molecules are usually present in the colloidal state. It is these proteins which with water form the gluten which is so essential to satisfactory baking. *The colloidal character of the proteins determines the main baking properties of the flour.* It is here that the *pH* concentration of approximately 5 is of such great importance. Great stress is laid on maintaining certain air conditions in the bakery during the making of bread, but according to investigations that have been made not only in this country but in Europe, a slight variation in the *pH* concentration of the dough will affect the quality of the bread and its possibilities for retaining moisture, infinitely more than any variation in atmospheric conditions.

To entertain hope, however, of developing a solution for the need of air conditioning in the bakery or its proper application, without a description or a definition of the colloidal state, is futile. In defense of this contention, one of the great English chemists, in an article on colloidal behavior in breadmaking has stated "that no manufacturer is less aware of the chemical problems underlying his trade than the master baker." If a chemist, working almost entirely in cereal chemistry may make this statement of the master baker, the writer feels that he may also state that the air-conditioning engineer is almost equally unaware of the chemical and physical problems underlying his profession, and unless the colloidal state of the ingredients entering into the manufacture of bread are understood by the air-conditioning engineer, he can never hope to solve the problem of air conditioning in the bakery.

The chemist who has made the statement about the master bakers has said that there has been no stimulus for master bakers to learn the chemistry of their trade, since in the bakery the feeling, tasting or smelling of the ingredients are better guides in the making of bread than the use of test tubes and balances. It is, however, the desire to obtain the greatest yield from the ingredients that has eventually turned the large producer of bread over to the chemist. It is, therefore, from the economical standpoint that the chemist

has come into his own in the bakery, and it may be from the economical standpoint that the scientist and physicist may come to their own in air conditioning.

Briefly, all substances when in solution behave either as crystalloids or colloids. If a substance in solution is present in molecular or ionic dispersion the solution is called a true solution. The size of such particles does not exceed one millimicron. If a substance in solution forms aggregates of greater than molecular size or if the molecules are very large, such as for instance, the protein molecule, the solution is called a colloidal solution and the substance may be said to behave as a colloid. The size of such aggregates varies between one millimicron to 100 millimicrons. If the size of the aggregates is still larger, so as to exceed 100 millimicrons and to be visible under the microscope, a mechanical dispersion results.

Substances in colloidal solutions may again be divided into two main groups, lyophilic dispersions and lyophobic dispersions. Lyophilic dispersions are solutions in which the dispersed particle solvates, that is, absorbs the solvent. If the solvent is water, the process may be termed *hydration.* Proteins, gums and starches belong to the lyophilic colloids. Most of the substances cannot be obtained in crystalline form, although egg albumen and haemoglobin have been crystallized. Lyophobic dispersions are colloidal solutions in which the dispersed particle does not absorb the solvent. Most of the inorganic sols belong to this class.

A substance colloidal in character will not dialyse, that is, it will not pass through a semi-permeable membrane. Semi-permeable membranes are of the utmost variety, the most familiar example is a colloidion membrane. A fish bladder is another such membrane. Substances in true solution readily pass through such a membrane while a substance in colloidal solution will be held back. Dialysis is, therefore, a method by which true solutions may be separated from colloidal solutions.

Substances can be obtained in the colloidal state in two main ways; (1) by taking them in molecular solution, that is, a high degree of dispersion, and then partially coagulating them by special methods; or (2) by taking the substance from the coarse suspension end of the scale, and increasing by electrical or mechanical means their degree of dispersion, so that they come within the limits of size and act or are colloids. From the miller's standpoint, the important *colloid* is protein. Proteins are lyophilic colloids, they solvate and their power to absorb water and swell depends upon this process. The *pH* of the solution is the determining factor of the extent to which solvation will take place. Proteins are ampholytes, that is to say, they behave either as bases or acids depending upon the *pH* of the solvent.

The point at which the protein sends out equal numbers of anions and cations and does not migrate in either direction if placed in an electric field is called its iso-electric point. At this point the protein is least reactive, has its lowest swelling capacity, has minimum conductivity and is most easily precipitated out. The iso-electric point of proteins is determined by measuring the *pH* of the solution at which the protein shows least swelling, conductivity, and migration in the electric field, that is, electrophoresis. The iso-electric point varies greatly for the different proteins, thus for gliadin it is at *pH* 10.2, for edestin, 7.9, while for egg albumen it lies at *pH* 4.8.

Proteins are good adsorbents as are all substances in the colloidal state.

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Adsorption is a surface phenomenon and the degree of adsorption is determined by the surface area and the electric charge of the particles. The surface area of particles in colloidal solution is very great. The protective action which certain substances in the colloidal state will exert upon another substance in colloidal solution depends upon the adsorbility of the protecting colloid on the dispersed particles. The coagulation of substances in colloidal solution upon the addition of another colloidal solution depends largely upon mutual neutralization. In both these phenomena, the *pH* of the solution thus plays a major part.

Flour is essentially a mixture of starch, electrolytes, moisture and dried hydrogel known as gluten. On the power of this gluten to swell, that is, on its power of solvation, or in flour, hydration, depends its useful qualities. This swelling is partly under control and partly dependent on the original coagulation in the sap juices during the ripening stages of the wheat.

Coagulation can be brought about if the particles are negatively charged by the addition of hydrogen-ions which are positively charged; in other words, by the addition of acids. Final coagulation is probably electrical, but the coagulation of an emulsoid is essentially due to the de-hydration action. Colloidal solutions are termed suspensoids, if the particles are solids. If the particles are liquids, they are called emulsoids.

In the swelling of colloids, which takes place in the making of dough and the baking of bread, there are usually large energy changes; and heat, often in a considerable amount, will be given out. Also, gels which include hydrated doughs, often separate out into two phases, one of which is a liquid. This liquid is not, however, pure water or pure solvent, but a colloidal solution in both the colloidal and molecular degrees of dispersion. This reaction is the reverse of swelling, and the staling of bread is said to be due to this separation.

It is interesting, at this point, to mention the first account of commercial colloidal chemical reaction. In the Bible, when the Israelites were bidden to make bricks without straw, the Egyptians, without possibly knowing the colloidal chemical reason for the reaction of straw, knew from experience that the straw was necessary, if the bricks were to be strong and uniform in structure. It is possible that the colloid of the straw acted as a protective colloid to the colloid of the clay, and prevented coagulation, which would have caused the materials to become irregular, non-homogeneous and unreliable. This same method of reasoning has since been used in industrial developments in the ceramic field.

The definition and discussion of hydrogen-ion concentration and colloidal behavior of starch or gluten has been entered into for two reasons. *First*, because the adsorption of moisture by the flour or the gluten in the flour for the production of a proper loaf is dependent upon the hydrogen-ion concentration and upon the colloidal state of the resultant doughs. *Second*, because both of these conditions, that is, the hydrogen-ion concentration and the adsorption due to the colloidal nature of the ingredients are distinctly electrical, and very large forces are essential to the disturbing of the equilibrium of the dough or the loaf. Certainly the changes in the hydration of the dough or the loaf, are not due to the common conception of the effect of regain due to air conditioning, since it is almost impossible, except under the attack of very high temperature, applied for long periods of time, to remove more than a

very small percentage of the hydrated solvent, and then only at the expense of changing the entire structure of the ingredients themselves. As already stated, even the staling of bread, usually considered a dehydration factor, has been proved by innumerable tests to involve only a very small loss of moisture, and a very definite change in the chemical structure of the ingredients.

It has been proved that the variation in cohesent water contents of gluten extracted from different flours is due to the development of electric changes around the particles of the protein or to chemical inter-action between protein acid or alkali and water, and that the density and water content of a solid mass of moist gluten depends, to a certain extent, on the disappearance of the electric double layers surrounding each particle.

FERMENTATION

In the fermentation process, the colloidal nature of the yeast or the enzymes affected by the yeast is greatly affected by temperatures. Jago, in his comprehensive study of the technology of bread-making, has stated that success in the manufacture of bread (sound materials, of course, being a first requirement) like all other operations in which fermentation is important, is dependent upon a proper regulation of *temperature*, because extremes of temperature in either direction weaken the action of yeast. In order to insure a uniform dough, it is essential that temperatures be kept as nearly constant as possible, during the whole of the fermentation.

Good yeast works well at from 80 to 85 F and at these temperatures lactic and butyric fermentation proceed slowly, even in the presence of those organisms which induce these types of fermentation. Sometimes cold is most detrimental and may even produce a sour loaf. However, to return to the subject of fermentation, the expanding of the dough so that it may hold the water of hydration is due to the expansion of the colloidal gluten cells by means of the expansion of carbon-dioxide gas, which eventually is released or forced out by its undeveloped pressure, partly during the fermentation period, and partly during the baking period. This formation of gas to which the leavening of the bread is due, is a very interesting chemical reaction, and is due to the catalyzing effect of living organisms or yeast spores on the carbohydrates inherent in the flour itself. These living organisms so familiar in all chemical changes in organic chemistry, have the effect of decomposing the starches into sugars, that is, of inverting cane sugar eventually into glucose and from glucose into carbon-dioxide and alcohol.

As previously stated, the *pH* concentration of the yeast affects materially the amount of carbon-dioxide that is developed from the sugars in the starch, and from the additional sugars added in the doughs. The maximum is the same *pH* concentration that the gluten reacts best to, that is *pH*5. Pasteur has called this *the correlative phenomenon of vitalax* and it may be applied to that group of chemical changes which are related to the life or developments of certain minute microscopic organisms.

The term fermentation in its widest sense refers to the microbiological transformation of carbohydrates, proteins and fats into their simpler components. If this decomposition is brought about by chemical means, strong acids or alkali and heat are necessary. At the temperature and *pH* at which most fermenta-

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TABLE 1. VARIOUS DATA OF DOUGHS

	1	2	3	4	5	6	7	8
Solids of doughs calculated from weights taken, per cent	57.44	57.38	57.00	57.30	57.30	57.30	57.15	57.30
Water of doughs calculated from weights taken, per cent	42.56	42.62	43.00	42.70	42.70	42.70	42.85	42.70
Solids determined on new-made dough, per cent	57.28	57.79	....	....	57.50	57.50	57.66	58.24
Water determined on new-made dough, per cent	42.72	42.21	....	....	42.50	42.50	42.34	41.76
Loss of dough in making, in grams	1.0	2.0	2.5	2.8	1.4	2.7	3.5	3.9
Dough taken for fermentation, in grams	292.0	294.0	297.3	297.7	296.1	297.7	296.9	299.0
Loss of weight during fermentation, grams	0.40	1.50	4.00	4.60	3.80	3.20	5.40	5.60
Volume of gas evolved at 64.4 F and 760 mm pressure, in cc	0.00	0.00	1,670	2,609	1,820	1,827	2,387	1,836
Weight of gas, CO_2, evolved in grams	0.00	0.00	3.09	4.71	3.37	3.38	4.42	3.39
Approximate weight of sugar required for production of gas, in grams	0.00	0.00	6.68	10.40	7.28	7.31	9.55	7.34
Approximate weight of alcohol produced, in grams	0.00	0.00	3.34	5.20	3.64	3.65	4.77	3.67
Weight of alcohol produced per 100 grams of dough, i.e., per cent	0.00	0.00	1.12	1.76	1.24	1.22	1.65	1.22
Loss of weight during fermentation, per cent	0.13	0.51	1.34	1.56	1.28	1.07	1.82	1.87
Sum of the two preceding quantities	0.13	0.51	2.46	3.32	2.52	2.29	3.47	3.09
Weight of sugar required per 100 grams of dough, i.e., per cent	0.00	0.00	2.24	3.53	2.48	2.45	3.21	2.45
Difference between the two preceding quantities	0.13	0.51	0.22	–0.21	0.04	–0.16	0.26	0.64
Soluble extract of the fermented dough, per cent	4.07	5.91	3.81	3.97	3.50	4.28	4.63	4.94
Subtract soluble matter of unfermented dough, per cent	3.29	3.33	3.29	3.29	3.89	3.92	3.28	3.91
	0.78	2.58	0.52	0.68	–0.39	0.36	1.35	1.03
Add sugar required for production of gas, per cent	0.00	0.00	2.24	3.53	2.48	2.45	3.21	2.45
Soluble matter produced during fermentation of dough, per cent	0.78	2.58	2.76	4.21	2.09	2.81	4.56	3.48
Ditto calculated on water-free solids of dough, per cent	1.35	4.41	4.88	7.58	3.76	4.97	8.21	6.12

tions proceed, the substrate (substance acted upon by the ferment or enzyme) is stable. Fermentation, when first discovered, was thought to be a microbiological action, but it has been shown that extracts of the organism were also capable of bringing about the decomposition of the substrate. Thus, alcoholic fermentation can be brought about either by living yeast cells, or by yeast extract free from any cells, or dead yeast cells.

Yeast is rich in enzymes. Diastase, the starch splitting enzyme, invertase, the enzyme which breaks down sucrose to its two component hexoses, glucose and fructose, and zymase, the enzyme which produces alcoholic fermentation of the hexoses are all present in yeast. In alcoholic fermentation the hexoses (glucose, fructose or maltose) are broken down to CO_2 and ethyl alcohol, C_2H_5OH, the theoretical yield being two molecules of ethyl alcohol and two molecules of CO_2 per molecule of hexose.

As stated, temperature is of great importance in influencing yeast activity. The temperature most favorable for activity as well as growth is from 77 to 95 F. At a temperature lower than 77 F, yeast growth proceeds, but slowly. At a temperature of approximately 50 F, the action of the yeast is arrested. However, the vitality is not destroyed, but only suspended, for at a higher temperature it again acquires the power of fermentation. Even freezing does not destroy yeast, unless the cells are injured. Above 95 F heat tends to weaken the action of yeast, until at 140 F the yeast is destroyed and it is at this temperature that the protein begins to coagulate. Of course, when dried, the cells can stand higher temperatures than when suffused with water. The whole theory of the formation of carbon-dioxide by the action of yeast on the various sugars in doughs is of extreme interest as paralleling to a great extent, the action of enzymes on the digestive system of the human body.

As affecting the value of air conditioning in the bakery, it is interesting to note that the loss during fermentation amounts to from 3 to 6 per cent and dough allowed to stand from 8 to 9 hours at a temperature of 85 to 90 F and then maintained at a constant temperature of 212 F for ten days, was found to have lost only 2½ per cent compared with the flour, and after the extreme temperatures of baking of from 450 to 500 F, the baked bread contained from 35 to 40 per cent of moisture.

Table 1 which is taken from Jago's Technology of Bread Making would indicate that the loss during the fermentation period is approximately the loss due to the evolvement of carbon-dioxide. The excess in weight lost over the amount of gas evolved is from 1 to 1.85 per cent. Therefore, very little escapes from the dough during fermentation, except the gas produced, the alcohol remaining in the dough, and these two weights, that is the carbon-dioxide and the alcohol approximate very closely the amount of sugar which has disappeared. A certain amount of soluble matter is produced by fermentation and this may be driven off. However, this is not likely to be affected by exterior conditions.

GENERAL USE OF AIR CONDITIONING

There is practically no mention in any of the articles on any investigations of bread-making of any definite effect of moisture content of the surrounding atmosphere on the quality of bread, whereas temperature and drafts are of

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continual recurrence in all of the technology on the subject. However, as has been stated, a definite condition of temperature and humidity is considered essential to proper machine manufacture of standard loaves of bread. In Fig. 1 the conditions considered most favorable to the making of bread have been indicated. There is no doubt that the relative humidities in the various departments of the machine bakery are of definite importance, and have a definite effect on the flour, the flour itself being only slightly colloidal. The hydroscopic nature of a cellulose material is affected by a difference in relative humidity but not to anywhere near the extent ordinarily imagined. In fact, if the air-conditioning engineer were dependent upon the conditioning of the flour storage for the output of his product, he would find himself materially lacking

FIG. 2. DOUGH ROOM OF BAKERY SHOWING DOUGH TROUGHS AND AIR DISTRIBUTION PIPING

in work, because probably in not one out of a thousand bakeries would he ever install air conditioning in the flour storage, and the slight difference in moisture content of the flour due to dry or moist conditions of storage space seldom exceeds 2 per cent as already pointed out. The probabilities are that air conditioning in the flour storage, affects much more, the development of bacteria, such as molds, than it does the moisture of the flour.

From the flour storage to the fermentation room, there is no question that temperature is the determining factor, and this temperature is maintained solely with the idea of regulating the rate of fermentation. In other words, as fermentation should not proceed while the ingredients entering into the dough are being homogenized, means are required to offset the heat of the mixing processes in order that homogenization should proceed to its maximum point before the yeast becomes active.

During the fermentation period, when the doughs are allowed to stand in troughs in a specially constructed room (see Fig. 2), a temperature of about 80 F, which is satisfactory to the fermentation process, should undoubtedly

be maintained and maintained at a constant point during the 3 to 5-hr period of fermentation.

There is no doubt that the passage of air over the exposed surface of the dough has a tendency to create a crust and it may possibly be that this crust is due to a difference in potential between the air passing over the ionized particles and the potential of the ionized air, and that the moisture content of the air required with the method of introducing this air for the maintenance of a definite temperature must be accomplished at such a high relative humidity that no static discharge is possible. This, to a certain extent, has been proved by one of the large bread manufacturers, for with a temperature of 80 F maintained in practically still air a much lower relative humidity than the 70 per cent usually called for by the air conditioning engineer has given satisfactory results. It is true that a certain amount of moisture is essential even under these still conditions, but very much less in quantity than has been regarded essential during the so-called air conditioning period of bakery development.

It is questionable whether the method of maintaining the desired temperature by direct radiation or by so-called bunker coils does not contain elements just as objectionable as the slight crusting which may occur due to the introduction of large quantities of air of the right condition. Furthermore, in the fermentation room where relative humidities of from 70 to 75 per cent are essential in order to prevent crusting, uniform conditions in the room are readily maintained and where all the air is handled through one central apparatus, the introduction of make-up air to take the place of air which has leaked outside is readily accomplished with modern air-conditioning equipment.

If this air could be introduced along the floor and under the troughs so that no air at high velocity impinged on the dough, probably lower relative humidities could be utilized than are now employed, with eminently satisfactory results. The crusting of dough is certainly to be avoided as it prevents an even release of carbon-dioxide and allows the formation of bubbles in the fermenting dough with consequent coagulation and unequal fermentation.

It has been stated that yeast is a living organism and is very greatly affected by change in its environment. Therefore, great care has always been essential in the handling of doughs from the time fermentation starts, and as doughs after fermentation are rather roughly handled in the make-up department, a certain amount of crusting is necessary in order to build up the gluten envelopes before entering into new processes or new fermentations. Therefore, the dry-proofing or rest period after dividing, scaling and rounding has never been one of the departments in which the air conditioning engineer has had any particular interest, as temperature alone was considered of importance in this particular department, but probably the same coagulation which takes place in the fermentation room itself occurs, although to a minor extent, in the doughs entering the rest period and this could be avoided if uniform conditions for the prevention of unequal crust hardness were utilized.

However, after the dry-proofing, when the rounded doughs in their respective containers are subjected to final proofing, air conditioning is in its *happy hunting grounds.* In other words, practically all American machine-made bread today is subjected to a final proofing in a special box in which a definite condition of temperature and humidity has been maintained with

entirely satisfactory results. An illustration of a modern proof box is shown in Fig. 3.

It is almost a universal practice to maintain a temperature of 95 F during the final proofing period. This temperature is consistent with all of the technology of bread-making, in that it corresponds to the maximum development or maximum satisfactory development of the yeast. The 70 per cent relative humidity, which it is customary to maintain in this final proofing, cannot possibly have any effect except on the exposed surface of the dough in the pan.

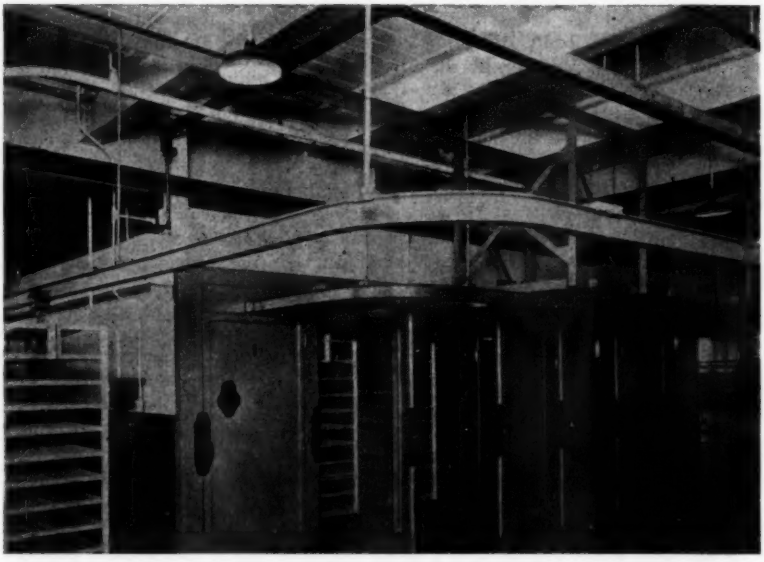


FIG. 3. FINAL PROOFER SHOWING RACKS FOR HOLDING BREAD SUPPORTED ON OVERHEAD TROLLEYS

Whether the fact that the relative humidity required for proper final proofing is the same as the relative humidity required in the fermentation room, under conditions previously mentioned, is due to an electrical phenomenon or not is a matter that should be thoroughly investigated. Certainly, the maintaining of a definite condition in the final proofer allows all of the pans to develop in the same length of time, and consequently permits of the removal of the fully developed dough at a very definite period, which corresponds with the time schedule of the balance of the bakery.

It is very doubtful whether the percentage of moisture in the air has any particular effect on the ultimate weight of the baked loaf. Certainly, the final proofer, when the conditions are recorded on recording instruments, is an efficient check-up for the manufacture of the loaf. However, it is the writer's opinion that the introduction of moisture-laden air at a definite temperature into the final proofer is much more a temperature problem than it

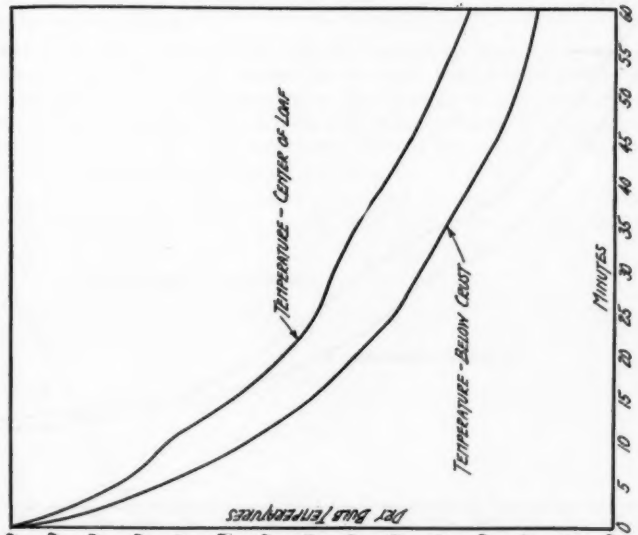


FIG. 4. COOLING CURVES FOR 1½ LB. VITAMINE BREAD—ROOM TEMPERATURE 79 F

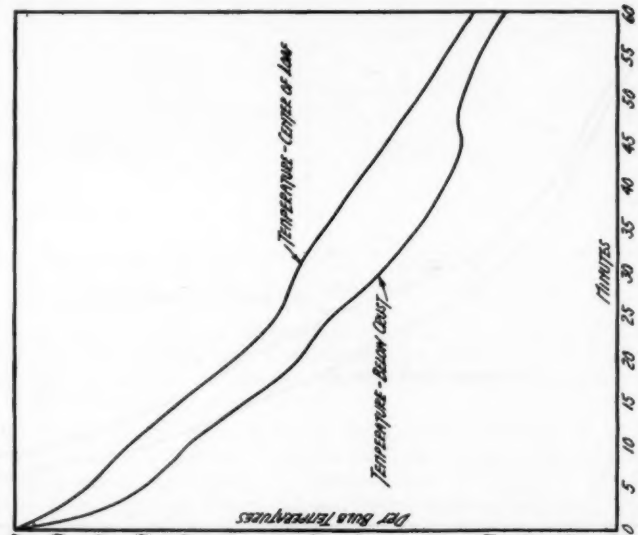
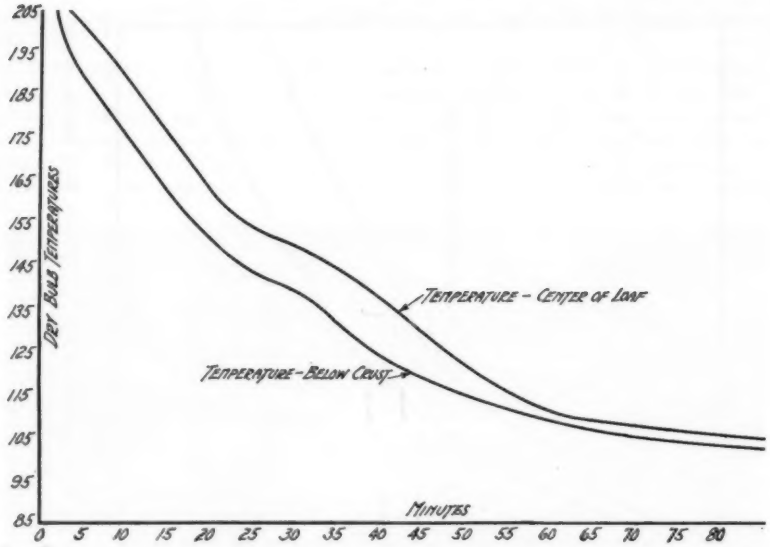
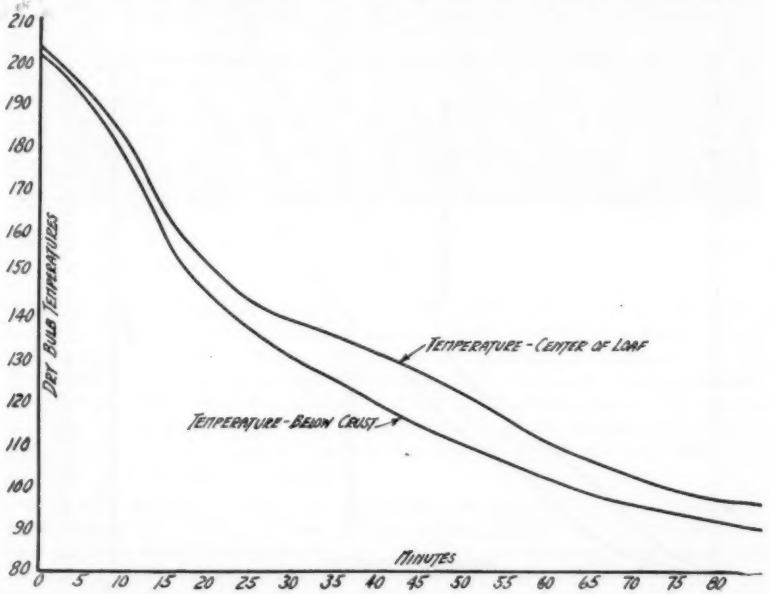


FIG. 5. COOLING CURVES FOR 1½ LB. SOFT BUN BREAD—ROOM TEMPERATURE 80 F





FIGS. 6 AND 7. COOLING CURVES FOR BAKERS' BREAD

is a moisture problem, unless the high percentage of moisture affects the hydrogen-ion concentration.

From the final proofer, the pans are introduced into the oven, and in this particular case there is no doubt that the high temperature of the oven is for the coagulation or caramelization or sterilization of the baked loaf, and the introduction of steam into the oven is certainly done for no other purpose than to assist in this caramelization, the moisture and temperature of the steam caramelizing the sugars on the surface.

The loss of weight which occurs in the oven is principally due to the moisture driven off from the surface of the loaf and from the volatiles which are

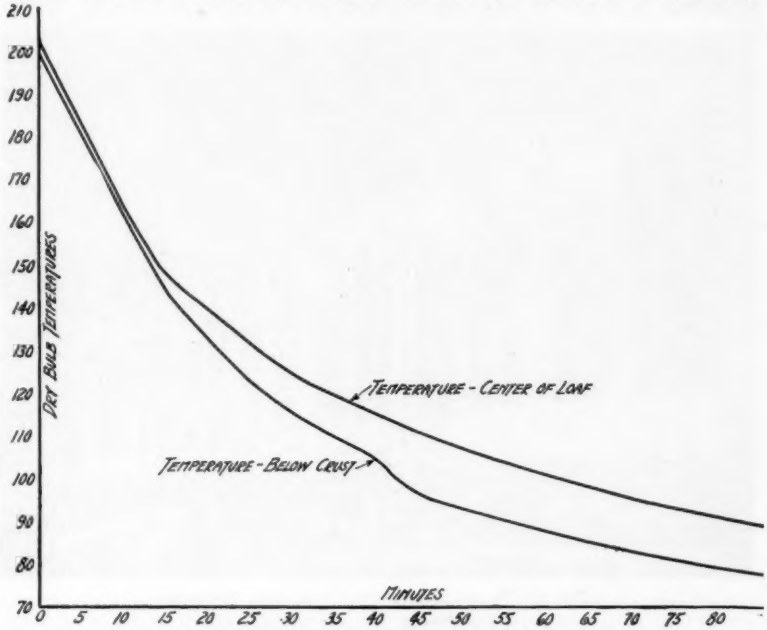


FIG. 8. COOLING CURVES FOR BAKERS' BREAD

released in the oven, and such slight amounts of carbon-dioxide as remain after the final proofing and from the alcohol which is a by-product of the fermentation process, the alcohol remaining in the loaf until driven off by the temperature of the oven.

When the bread which is finally baked and is encased in a practically impervious crust is removed from the oven at a temperature of about 200 F, cooling proceeds very rapidly as shown in Figs. 4, 5, 6, 7 and 8. Figs. 9 and 10 show views of bread coolers of two large western bakeries. The amount of moisture lost during the cooling process is slight, probably not over 0.5 per cent, which percentage would correspond to the cooling effect due to evaporation of the moisture and volatiles involved in the cooling process.

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There is considerable controversy as to the reason for the necessity of cooling a loaf to a definite temperature, usually below 100 F before hermetically sealing it in some exterior wrapping. It is the opinion of the writer that temperature is of considerably more importance than humidity and possibly the temperature has a great deal to do with the development or formation of mold growth and the keeping quality of the bread. In other words, as all ferments are less active at low temperatures, and as bacteria are in the nature of true ferments, the cooler the loaf, the more unlikely the formation of any deleterious growth.

As stated earlier in this paper, even the staling of bread which was assumed to be due to a loss in moisture has been proved to be a chemical change

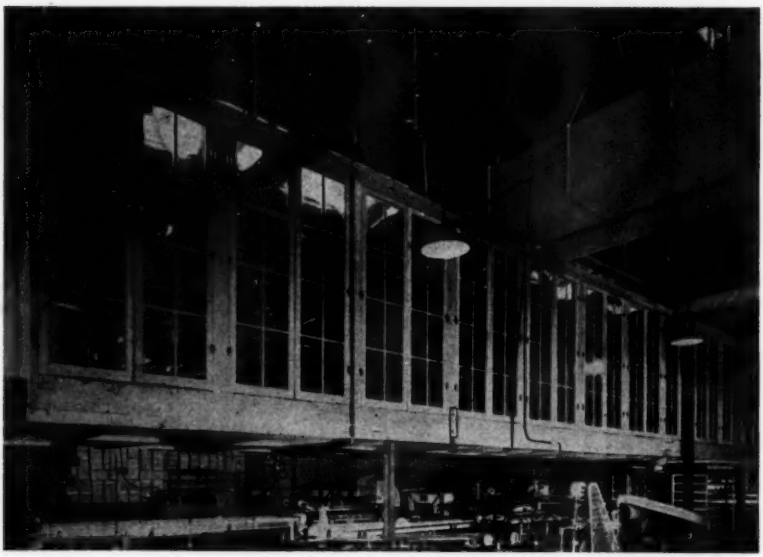

FIG. 9. BREAD COOLER IN A WESTERN BAKERY

and not a dehydration process. Therefore, the suggested vacuum method of removing heat and moisture from baked loaves does not seem absolutely logical when the chemical changes which create the stale or inferior loaf are taken into consideration.

CONCLUSIONS

The supreme importance of the hydrogen-ion concentration of the various ingredients that enter into the making of bread; the relationship between the hydrogen-ion concentration and the colloidal nature of the principal ingredients of bread, which colloidal condition is itself electrical; the possible hydrogen-ion concentration in air at 70 per cent relative humidity which has been over a course of years, found most effective in the present method of conditioning doughs and bread, through the bakery—these interesting problems should be

investigated further with respect to the use of air conditioning as such, in the making and baking of bread. An investigation is suggested of the strangely familiar development of bread structure, a colloidal state; and also of the maximum temperature of efficient fermentation affecting the development of all articles of food with the necessity of the introduction of food into the digestive system in a colloidal state, and the most efficient temperature of life of approximately 98 F. Whether an electrical explanation of the various phenomena entering in to the development of food and its digestive qualities

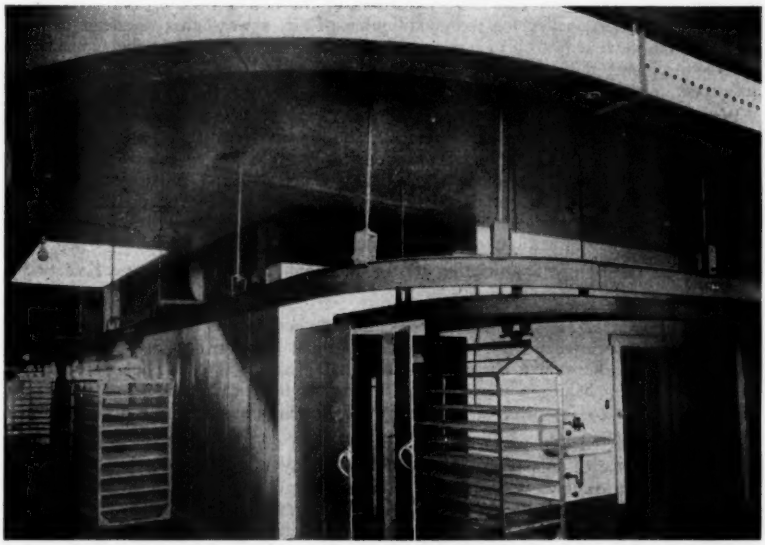

FIG. 10. BREAD COOLER WITH AIR CONDITIONING UNIT ON TOP SHOWING BREAD RACKS SUPPORTED ON MONO-RAIL TRACK

and the hydrogen-ion concentration of the blood at different relative humidities will give rise to new concepts of life and health are still to be developed in the physical laboratory.

DISCUSSION

CHAIRMAN W. H. CARRIER: It is papers of this type, that will be ultimately useful in arriving at some of the facts regarding certain applications of our art. Discussion is in order.

R. P. BOLTON: This scholarly paper, apart from the information which it contains is one that opens the door to further investigation upon the same line.

The future of our Society and the development of our art lie along the lines of close investigation of cause and effect. We are working today upon results that have been stated for us by experiences of the past merely as physical results. Those things we know and find in our textbooks. But today we must go further into those

subjects. We are heating engineers, and we may ask ourselves, "What is heating? What is this transmission of heat? What is going on in our own bodies as we receive this heat from the heating agents that we employ?" Those are the subjects that we will have to investigate in the future, and the chemist must help us, as well as the physicist.

This paper marks a step in the progress of the art of the heating and ventilating engineer, and we owe a debt of gratitude to Mr. Fleisher for having given us so freely the facts and information that surround a branch of the work in which we are engaged, with which few are familiar. I hope that this may be followed by papers that will deal with other less known features of the art of heating and ventilation. The door has been opened for the conditioning of air, a word that does not express the full value of that branch of our art, but when we have arrived at that point these are the problems that will have to be solved, and questions that will have to be answered.

We must go further into the details of what happens to us, and to the materials which are dealt with when we condition air, and we must remember, as Mr. Fleisher said, that after all the condition of the air is probably affecting only our own surfaces. What goes on inside of us we haven't yet determined. What goes on inside of some of the brains of some people may be like the dough that Mr. Fleisher described. It may be compounded of interesting chemical compositions, and subjected to curious combinations of physical changes.

I wish to record my appreciation of the information afforded to our Society by Mr. Fleisher.

CHAIRMAN CARRIER: The problem of colloidal chemistry is one of the most intricate and abstruse in chemistry. I wish that we had more in our Society to discuss this subject. Perhaps in a chemical society that phase of it could be done, but we can place the emphasis on the air conditioning angle, and Mr. Fleisher has given the chemical basis, I assume, of reasons for air conditioning requirements.

MR. FLEISHER: In response to various comments which have been made relative to the significance of this paper, I should like to say that the bakery was taken as an example of a complex subject which, due to the amount of chemical study which had been made, was susceptible of analysis. The subtitle of my article should have been The Change in Moisture Content of Most Colloidal Materials or Conglomerates Is Due to a Change in the Iso-electric State of the Material through a Differential in Electrical Potential between the Material and the Potential of the Surrounding Atmosphere.

The particular activity of the product for its best condition is a function of the *pH* concentration. There is a particular *pH* concentration of the surrounding atmosphere which lends itself best to the maintenance of that particular condition in the material.

A change in an electrical condition can only be brought about by superimposing an electrical force. Consequently any changes that do take place must necessarily be electrical basically.

In the realm of comfort the electrical equilibrium of our bodies, both superficially and internally, can only be changed by an electrical impulse contacting with them. The ionization of the air, translating heat factors to electrical factors or ionized factors, are the basic energies which check or change our metabolism to create comfort or discomfort.

Just as flour, gluten, yeast and doughs can be analyzed and as equations between their electrical forces and the surrounding electrical forces, both the forces known as heat and the forces known as electricity or ionization, could give the best results, so this same method can be applied to many other products such as paper, ceramics, tobacco, and finally, the most important of all, to human beings. It is the function of the mathematician and the physicist to establish this mathematical balance.

No. 887

AIR CONDITIONING FOR RAILWAY PASSENGER CARS

By A. H. CANDEE,[1] EAST PITTSBURGH, PA.
MEMBER

AIR CONDITIONING for railway passenger cars presents intriguing possibilities to the heating and ventilating engineer as well as to the refrigerating engineer. While the needs of this field have been known for some time, it was not until 1929 that a commercial solution of the problem was undertaken. At this time the Baltimore and Ohio Railroad applied to one of their standard coaches means for cleaning and circulating the air within the coach, which also included a mechanical refrigeration system for cooling the air. The success of this trial installation led to the application of a modified and improved system to one of their colonial-type diners. This diner,[2] the Martha Washington, was placed in regular service in April, 1930, and was the first practical and commercial installation of its kind in the world. It was not actually the first air-cooled car in the world, however, for it has been the practice to cool cars in Egypt and India by hanging wet cloths or matting over windows and on the roofs to reduce the temperature within the cars by the evaporation of the water.

The operation of the Baltimore and Ohio car has been an outstanding success, and has attracted widespread interest among the railroads. The Santa Fe quickly followed with a similar dining car,[3] and other railroads will undoubtedly apply these principles in the near future. One railroad has also instituted the practice of pre-cooling passenger cars of some of its crack trains in hot weather by placing ice on a special truck and drawing air over this ice through the cars before the passengers are admitted. The details of the Baltimore and Ohio and Santa Fe installations are shown in Figs. 1 to 14.

It is recognized that the air cooling and conditioning of most of the railway passenger equipment is inevitable, once it has been shown to be practicable. Those who have had occasion to travel on the railroads in the summer time or who have entered a sleeping car which has been occupied for some time will appreciate the comfort of clean, fresh air at the right temperature. The

[1] Railway Engineer, Westinghouse Electric and Manufacturing Company.
[2] Railway Age, August 9, 1930.
[3] Railway Age, August 23, 1930.
Presented at the 37th Annual Meeting of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, Pittsburgh, Pa., January, 1931.

railroads also recognize the importance of this as a means of attracting the traveling public, which is showing a marked tendency to travel by automobile, bus, or airplane rather than by rail.

EXTENT OF AIR CONDITIONING FIELD FOR RAILWAY CARS

There are approximately 65,000 railroad cars in passenger service in the

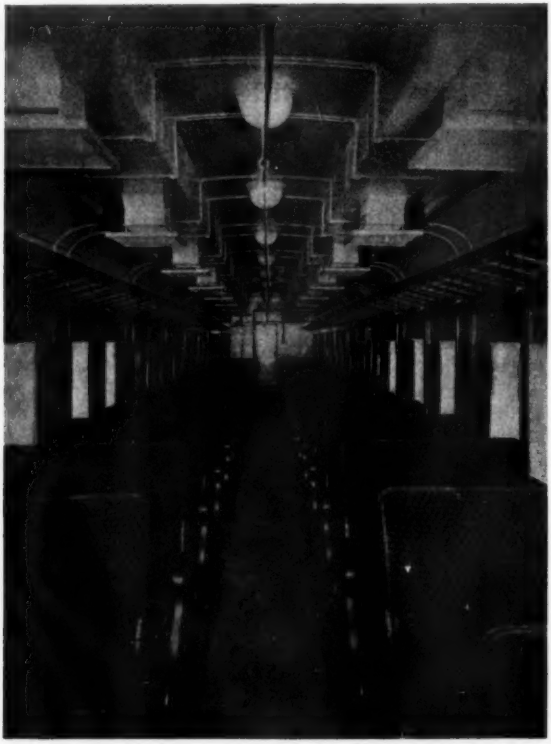

FIG. 1. PIONEER AIR CONDITIONING INSTALLATION IN A COACH OF THE BALTIMORE AND OHIO RAILROAD

United States, of which approximately 9,000 are Pullmans. The percentage of that number which may be considered as suitable for air conditioning is unknown, but a survey shows that there are approximately 165 named trains in operation which could be equipped advantageously with air conditioning apparatus. A conservative estimate of the number of passenger cars required to fill out these trains daily and protect the service is 5,500 cars. It is also estimated that there are at least 5,500 cars operating in other first class trains scattered throughout the country, upon which air conditioning apparatus may

be installed advantageously. This would then indicate the potential field for air conditioning to be at least 11,000 cars, and probably more nearly 15,000 cars if equipment is developed having low first cost and low operating expense.

The two air-conditioned cars in operation at present are in the nature of trial installations. They are demonstrating the feasibility of applying refrigerating plants to railways cars for cooling purposes, but the heating systems of the cars have been unchanged. The ultimate equipment will undoubtedly have the heating system modified at the time refrigerating equipment is applied,

TABLE 1. PREFERRED INTERIOR TEMPERATURE REGULATION
(*Exterior Temperatures above 70 F*)

External Temp. F	Interior		
	Dry-Bulb Temp. F	Wet-Bulb Temp. F	Effective Temp. F
95	80	65.2	73.4
90	78	64.5	72.2
85	76.5	64.0	71.1
80	75	63.5	70.2
75	73.5	63.0	69.3
70	72	62.5	68.2

so that the temperature of a car may be controlled at all times by the addition of heat to, or the extraction of heat from, the circulated ventilating air.

Air conditioning may be divided into three general phases:

1. Supply and circulation of cleaned fresh air.
2. Control of temperature of the air.
3. Humidity control.

For the immediate future, it is doubtful whether there will be any attempt to control closely the humidity of the air within a car because of the complications involved and the lack of space for equipment and water supply. The band of comfort to passengers is so broad that it is entirely feasible to adjust the air temperature to maintain comfortable conditions. Thus, in the winter time, a passenger will feel just as comfortable at a relative humidity of 10 per cent and an air temperature of 72 F as at a relative humidity of 50 per cent and an air temperature of 68 F. This means that while thermostatic temperature control will be used normally, it is advisable for the train crew to have some manual adjustment of all controlled temperatures within the cars of the train, to compensate for variations in atmospheric humidity. Such thermostatic control must permit of practically constant temperature during cold weather, but must allow of a rising interior car temperature as the exterior temperature rises, for the spread between the two should seldom exceed 15 deg (dry-bulb), or the effect on passengers is likely to be unpleasant. The preferable differentials in temperature with average external humidity are shown by Table 1.

BASIS OF AIR CONDITIONING RAILWAY CARS

One of the greatest advantages of any system of air conditioning is the provision of clean fresh air for the passengers. The opening of passenger car

windows is usually accompanied by a swirl of cinders, soot, and smoke, yet at present one cannot travel in the summer time without having them open. The basis of air conditioning must, therefore, be to have all windows closed and locked, and to draw the ventilating air through a cleaning device of some sort which will exclude the dirt and dust, after which it must be properly circulated within the car to provide equal distribution without excessive drafts. Sealed double-sash windows, improved insulation, and elimination of ventilators and bracket fans will no doubt be features of the air-conditioned car. This will also make a quieter car, as external noises will be more effectually excluded.

It is not possible to locate an air intake in a railway car where clean air from the outside is insured. It is, therefore, necessary to remove the cinders,

FIG. 2. BALTIMORE AND OHIO DINER

dust and dirt by artificial means, such as an air washer or a dry type of filter. While the former has the advantage of being a cooling medium and a means of controlling the humidity, the space required is a serious handicap in a great many cars. The dry type of filter will probably be used, if an air washer cannot be applied, and may even be used in series with the air washer to remove the larger particles from the air stream. Any dry type unit must be located for easy removal for cleaning purposes. It appears that the cleanest air may be obtained high in the car, near the roof. The two existing installations previously mentioned draw in air through the vertical walls of the clear-story just above the half deck of the car, where deck ventilators are normally placed.

AMOUNT OF AIR REQUIRED

The amount of air circulated per minute per passenger in a railway car equipped with air conditioning apparatus will vary somewhat with the particular type of car. It is clearly necessary to recirculate a portion of the air in order to reduce plant capacity required for heating or cooling purposes where limited power and space are available, and the amount of fresh air drawn from the exterior of the car will vary with atmospheric conditions and with the application. It is obvious that a dining car must have more fresh air supplied (on account of pollution due to kitchen smoke and odors) than a

TABLE 2. APPROXIMATE DATA FOR PASSENGER COMPARTMENTS

	Length Compartment (Feet)	Floor Area (Sq. Ft.)	Interior Volume (Cu. Ft.)	Maximum Seating Capacity	Volume per Passenger (Cu. Ft.)
Coach	58	522	4060	100	40
Sleeper	37½	337	2740	48	57
Parlor Car	46	414	3360	26	130
Diner	39	350	2850	36	80
Smoking Room	10	65	540	6	90
Drawing Room	7	43	380	5	76

parlor car, and that a smoker must have considerable fresh air in order to clarify the interior atmosphere. This indicates the desirability of selective variation in the ratio of incoming fresh air to circulated air. While railway ventilation practice in the past has generally been based on 15 cu ft of air per minute per passenger, it is expected that forced circulation and careful distribution of the air will reduce this figure somewhat. Table 2 is submitted as a guide in determining required air circulation for different passenger compartments. On the basis of 15 cu ft per minute per passenger, it would be necessary to change the air in a coach every 3 minutes, in a sleeper every 4 minutes, in a parlor car every 9 minutes, and in a diner every 5 minutes as a maximum. However, since air movement assists in improving the comfort of a cooled car, it is desirable to circulate the air at a faster rate than this in the summer time, reducing the fan speed and delivery for cold weather conditions.

From the foregoing discussion, it will be apparent that the essentials of the air circulating system include one or more air filters equipped with manually-operated shutters which may be adjusted at will, or completely closed in passing through tunnels, a circulating fan (preferably of variable manual speed control), and a suitable duct system for proper distribution of the air throughout the car and for recirculation of a portion of that air. The amount of air allowed to escape from the interior must be equal to that introduced, and with a sealed car will mean the provision of a variable vent.

LOCATION OF HEATING AND COOLING UNITS

With an air circulating system provided on a car, the best temperature control may be effected by placing both the heating and the cooling units directly in the ventilating air stream, preferably close to the fan, adding or sub-

TABLE 3. CALCULATION OF HEAT EXTRACTION FOR A SLEEPING CAR*

	Btu per Hour
Heat through car walls	11,000
Solar radiation	10,000
Heat from make-up air	30,000
Passenger radiation [300 Btu per hour per passenger (48 passenger maximum)]	14,400
Power of blower	1,500
Loss in pipe lines	2,000
Total	68,900

* Based on following conditions: External temperature, 95 F; internal temperature, 80 F; time, 2:00 to 4:00 P. M.; average train speed, 30 mph; maximum conditions.

tracting heat as required. The location of the heating coils in this way will improve the efficiency of heat transfer and save in cost and weight over the present system of extending steam pipes along the sides of a car near the floor. The fact that it may be necessary to circulate both the heated and the cooled air through the car in the same direction will probably be unimportant in the face of practical considerations.

Steam, generated at the locomotive, has long been the source of heat for maintaining comfortable temperatures within a train in cold weather, although

FIG. 3. INTERIOR VIEW OF BALTIMORE AND OHIO DINER SHOWING AIR OUTLETS UNDER THE HALF-DECK

this inflicts a severe handicap on motive power with low ambient temperatures. An ordinary coach having a 70-ft passenger compartment requires nearly 100 hp from the locomotive for heating under the maximum conditions in zero weather, which means that approximately half as much energy is used for heating as for train propulsion under these conditions. It is undesirable to draw power from the locomotive for auxiliary purposes if it can be avoided in any way, but the use of steam for heating cannot easily be discarded.

METHODS OF COOLING

For the cooling of cars in hot weather, means must be arranged for passing the ventilating air over a cold body in order to extract heat. There are three general systems available for providing such a cold body:

1. A material having heat-absorbing characteristics, such as ice.
2. An absorption system of mechanical refrigeration.
3. A compressor system of mechanical refrigeration.

Differentiation between the absorption system and the compressor system has been made on account of the fact that the one employs heat direct, while the other requires mechanical energy. In considering their application to railway cars, this may be of considerable importance.

Any refrigerating method should be carefully considered from a practical standpoint, weighing the relative effects of practicability, reliability, weight, first cost, and operating expense. It may be used a maximum of 50 days a

FIG. 4. CEILING LOUVRES FOR AIR RECIRCULATION IN THE BALTIMORE AND OHIO DINER

year, yet it must ride with the car for the other 315. The air circulating system, however, is used the year round.

The refrigerating system must be safe, above all else, allowing no escape of fumes within the car and no danger of explosion even in case of accident to the car. The weight, first cost, and operating expense are economic factors and must be justified on the basis of an increase in revenue, for the railroads cannot continue to provide additional comforts for the traveling public unless these comforts show returns in increased passenger traffic or unless the fare per passenger is increased.

The most common method of providing heat absorption capacity is by the use of ice. It is estimated that from one to two tons of ice will keep a car cool for at least a day under the most severe conditions if the melting is properly controlled. Such a method of refrigeration, however, requires daily recharging of the ice boxes, but compared with mechanical refrigeration, the apparatus carried on the car is simple.

Absorption systems are available for maintaining low temperatures in refrigerator cars, but the development has not yet progressed to a point where

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they may be applied to the air conditioning of passenger cars. The fact that heat is used in place of mechanical energy offers very attractive possibilities, as the fuel may be carried on the car itself, and may be replenished with little difficulty. It is expected that new developments in this field will be announced shortly, which will simplify the application of refrigerating systems to railway cars.

It is logical that the first air-conditioned cars should employ mechanical refrigeration, as this field has had the advantage of considerable development. The question of power supply, however, is troublesome and together with the refrigerating mechanism adds considerable to the weight and cost of a car.

FIG. 5. COMPRESSOR BOX OF THE BALTIMORE AND OHIO DINER

There are six general methods of providing the energy required for a mechanical refrigerating system (excluding the absorption system):

1. Steam power plant on each car.
2. Steam power plant on or near the locomotive.
3. Internal combustion engine power plant on each car.
4. Internal combustion engine power plant on the locomotive or in a baggage car.
5. Axle driven generator on each car.
6. Axle driven generator on each car supplemented by a storage battery.

It will be noted that in methods (1), (2), (5) and (6) the energy is obtained by subtraction from the motive power, except in cases where steam may be generated in a separate boiler such as is used in connection with an electric locomotive. Method (3) renders a car independent of movement or connected motive power. With method (5) energy is supplied only when the train is in motion and it is to overcome this handicap that method (6) was devised. The Baltimore and Ohio diner employed method (5), and the Santa Fe, method (6), but alterations in the former have been made so that the cooling water pump and the ventilating fans now operate from the battery and thus continue the cooling while the car is standing for short periods. Schemes such as (2) and (4) require that electric power circuits be carried the length of the train.

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FIG. 6. SIDE VIEW (KITCHEN END) OF THE SANTA FE DINER

While such a distribution system cannot be applied in the immediate future, due to the fact that it is an immense task to apply such conductors to the number of cars involved, it is obvious that the electrical comforts and conveniences such as are found in a great many homes will ultimately be available to the railroad traveler and that this will require a dependable power system. Some of the most prominent railway electrical engineers forecast the use of such power generation and distribution on trains, with the consequent elimination of the troublesome axle-driven generators and storage batteries now in use.

For the immediate future, therefore, it may be assumed that there are several practical methods of obtaining a cold body, namely, ice carried in an insulated

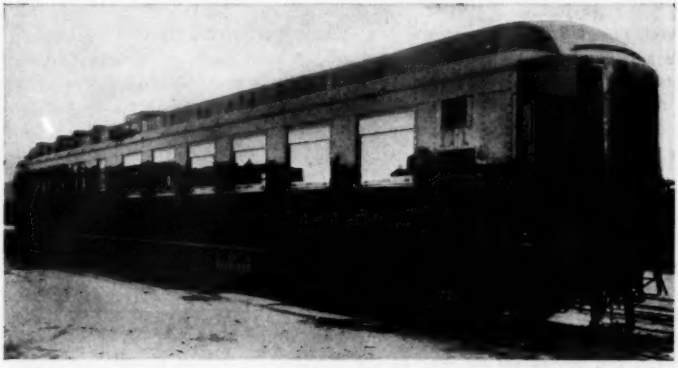

FIG. 7. VIEW OF THE COMPRESSOR SIDE OF THE SANTA FE DINER

compartment and melted as required; absorption systems of refrigeration using a stored liquid, gaseous or solid fuel supply which may be replenished at will; a steam or internal-combustion engine-driven mechanical refrigerating plant; or equipment using an axle generator as the means of utilizing motion of the car for transmitting energy from the locomotive. Although the two diners now in operation utilize axle generators, it is doubtful whether this practice will continue.

Calculations indicate that from 60,000 to 80,000 Btu per hour maximum must be absorbed in order to provide effective cooling of a diner or a sleeping car. Experience has shown that this requires an input of from 10 to 13 hp to a refrigerating plant of the mechanical type. The heat to be extracted is calculated as shown by Table 3, making due allowance for im-

FIG. 8. DINING COMPARTMENT OF THE SANTA FE DINER. NOTE THAT AIR OUTLETS ARE INCONSPICUOUS.

provements in insulation of the car. Any mechanical system of cooling must have a capacity equal to the short time demand such as is indicated in Table 3, but the total capacity of any system using heat absorbing material, such as ice, may be based on an integration of the demands throughout the complete run of the car between recharging stations.

DESCRIPTION B & O DINER INSTALLATION

In the installation of the air-conditioning equipment in the diner of the Baltimore and Ohio Railroad, the cooling coil of a mechanical refrigerating plant is located in a compartment above the passage way from the dining room to the pantry and kitchen. This consists of finned pipes carrying cooled water. Air is drawn from the outside of the car through two filters, one being mounted on each side of the car above the half-deck, and is passed through the cooler on its way to the electrically-driven circulating fans. The air is carried along the two sides of the roof of the car in ducts, as may be

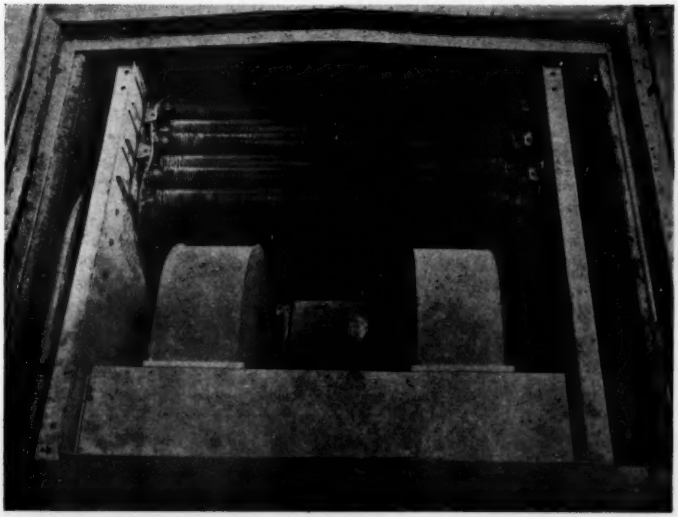

FIG. 9. AIR COOLING UNIT AND CIRCULATING FANS OF THE SANTA FE DINER

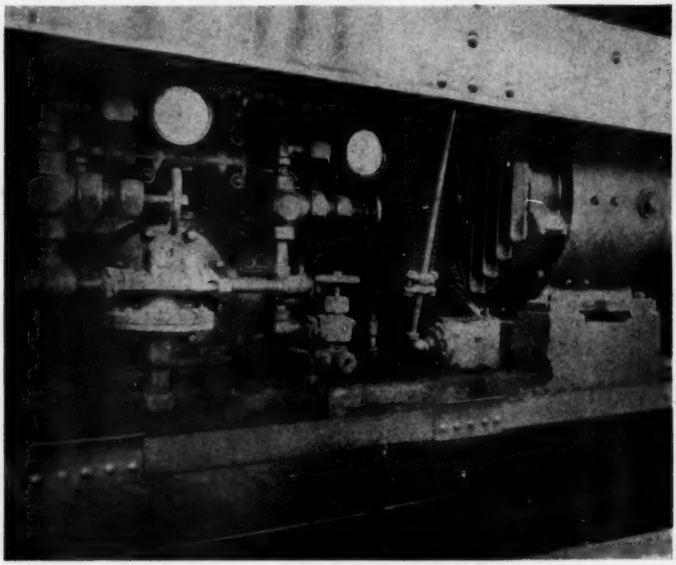

FIG. 10. MOTOR DRIVEN AMMONIA COMPRESSOR WITH FITTINGS OF THE SANTA FE DINER

FIG. 11. SPRAY TOWER FOR COOLING CONDENSING WATER OF THE SANTA FE DINER

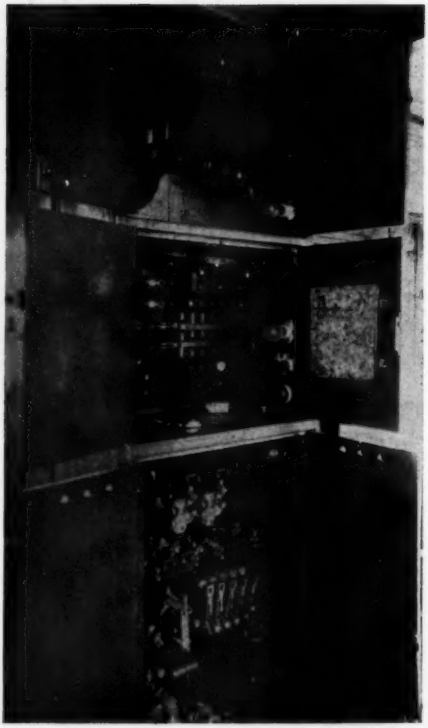

FIG. 12. CONTROL CABINET OF THE SANTA FE DINER

observed in Fig. 2, and enters the dining room through louvres distributed the length of the compartment, visible in Fig. 3. The discharge of air to compensate for that entering is effected through the kitchen exhaust fans and other doors which may be open. Two horizontal louvres in the bottom of the cooling compartment, shown by Fig. 4, permit of air recirculation and temperature control. One or the other of these is always open, either passing the recirculated air through the cooling coil or by-passing it to maintain temperatures. A thermostat governs the operation of the shutters over these louvres.

An ammonia refrigerating system is mounted under the car floor, a motor-driven compressor, a condenser, an expansion valve, and an evaporator com-

FIG. 13. ONE OF THE 7½-KW AXLE-DRIVEN GENERATORS OF THE SANTA FE DINER

prising the main equipment. The ammonia is cooled and condensed through the medium of water, which in turn is cooled in a spray tank occupying a space formerly used for a locker at one end of the car above the floor. Water is also used in the air cooler, this water being maintained at 40 to 50 F by passing through coils in the evaporator tank.

Circulation of water between the condenser and the spray tank and also between the evaporator tank and the air-cooling unit is by means of motor-driven centrifugal pumps. As previously stated, an axle-driven generator supplies the power for the compressor motor, the air-circulating fan, the motor in the spray tank, and the two water-circulating pumps. This generator, which is entirely separate from the axle generator which charges the storage battery and furnishes energy for lighting, normally develops 115 volts at a car speed of 28 mph and above, so that until the car reaches this speed, no energy is available for operation of the refrigeration system. The *cutout* voltage is approximately 80 as the train speed descends.

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DESCRIPTION OF SANTA FE DINER

The Santa Fe diner is similar in most respects to the Baltimore and Ohio diner. One of the differences is in the detail of air ducts and openings into the dining room, which may be noted by a comparison of the exterior and interior views. Another difference, of major importance, is that all of the power used for refrigerating, air circulation, lights and other purposes is generated by two axle generators at 32 volts (nominal) and a large storage battery is used in conjunction with these generators so that power is available for operation of the cooling system whether the car is in motion or standing.

FIG. 14. A PART OF THE 2250 AMPHR STORAGE BATTERY OF THE SANTA FE DINER.

The fact that the Santa Fe trains may experience extreme changes in temperature within a single day in passing from the deserts to the mountains makes a third difference necessary. The condenser and evaporator tanks are located on the roof along the side of the clearstory so that with the refrigerating system shut down, all water drains back inside of the car. These tanks may be observed in the side view of the car, Fig. 6.

The drive mechanism of an axle generator has always been troublesome, and storage-battery maintenance is a considerable factor in operation. The combination is essential if continuous power is to be provided from an axle-generator system. It appears to be more desirable to use a small internal combustion engine power plant on each car if vibration, noise and fumes may be rendered unnoticeable within the car. Individual steam power plants are probably not practical on account of the difficulty of supplying steam at reasonably constant pressure in the summer time when a car is in a yard or station without connected motive power.

PROBLEMS TO BE SOLVED

There are a great many problems to be solved. Who is now able to forecast the best method of refrigerating, of arranging units, of supplying energy,

and of insulating cars? These questions can be answered only by experience, and trial installations such as have been made have already advanced the available knowledge considerably. It is highly probable that the resulting research will materially reduce the amount of steam used for heating, through the use of double windows fastened in place and carefully sealed, through different materials and methods of blanketing the car, and by better heat transmission from the steam pipes to the air. This factor alone, if accomplished, will represent a considerable saving to the railroads.

Many railroads are interested in air conditioning, but few feel that the weight, cost, and complication of the present types of mechanical refrigerating units can be justified. It is one thing to apply such a system to a car, but it is an entirely different matter to train an organization, scattered over the country, in operation and maintenance so that the equipment will function continuously and efficiently. The problem is not alone that of the railroads, but involves the heating, ventilating and refrigerating engineers of the country, by whose efforts simple and reliable equipment can be developed. Preference will be given to those systems which employ the minimum number of working parts, the lowest operating expense, and the minimum of space requirements.

DISCUSSION

MR. RAGSDALE: The Santa Fe uses the same system for heating in winter as for cooling in summer and the M. K. & T. plans to do likewise. A Santa Fe engineer states that only in extremely cold weather has it been necessary to use ordinary heating coils around the base of the car, so as to heat the floor. Otherwise the system has proven adequate for ordinary car usage.

W. H. CARRIER: As designer of the first car, I want to correct some statements made by the author regarding the power required. I do not know the source of the figures he has given but I'd like to state that we found that theoretical calculations cannot be used in car design. There are too many variable factors such as, the effect of solar radiation and the air velocities over the car; but there are compensations there that can only be determined by experiment. The finest calculations in the world are useless without the practical application. It took one year of preparatory experiments to determine these requirements and no theoretical calculations can ever take the place of actual test loads.

MR. ADAMS: Mr. Chairman, are they too high or are they too low?

MR. CARRIER: Some are too high and others are too low, but the power requirements given are too high.

I think we are indebted to Mr. Candee for this paper. He has made a good analysis of the requirements in so far as the methods of operation are concerned. Theoretical calculations of the load, however, cannot be used successfully because, as I said, there are factors involved that are so variable under conditions of operation, that we found it was impossible to get any accurate estimate except in practice. I do not wish to criticize Mr. Candee's able analysis of the various means of power application. The subject is of much interest.

MR. CANDEE: I would like to suggest that if the cars do not have humidity control the figures may be "all wet," but the data are verified by the fact that a 10-kw axle generator was used in one case, and two 7.5-kw generators were used in the other case. That would correspond with 12 or 13 hp at least on the B & O car and approximately 20 hp in available generator capacity on the Santa Fe. The Santa

Fe, of course, used part of that energy for other electrical purposes on the car, but considering the fact that a 10-kw generator was applied on the B & O and that much of that capacity of the generator was used, it is fair to assume that in most cases at least 10 to 13 hp will be used. I agree with Mr. Carrier that the individual heat losses as calculated are subject to considerable variation.

No. 888

AIR LEAKAGE THROUGH VARIOUS FORMS OF BUILDING CONSTRUCTION

By F. C. Houghten,[a] *(MEMBER)*, Carl Gutberlet,[b] *(NON-MEMBER)*, and C. A. Herbert,[c] *(NON-MEMBER)*, Pittsburgh, Pa.

THE study of infiltration through masonry walls was undertaken at the Research Laboratory of the American Society of Heating and Ventilating Engineers in 1925. The tests were made in two pieces of apparatus (Fig. 1) designed for that purpose. Early in the investigation it was discovered that the rate of infiltration through masonry walls increased for several months after building, and as a result, the testing of different types of construction in the available apparatus in the Pittsburgh Laboratory became a very slow process.

In 1927 the major portion of the infiltration studies of the Laboratory was transferred to the University of Wisconsin under a cooperative agreement. Profiting by the earlier experience in Pittsburgh, the apparatus at Wisconsin was so designed that a large number of walls could be built in steel frames which could later be moved into the testing machine from time to time as the walls aged. This made possible a more rapid testing program, since the testing apparatus was not tied up by a single wall during the aging process.

Since 1926, the two pieces of apparatus at the Laboratory have been kept in constant use with the result that there has now accumulated data on several types of construction which are made the basis of this report. These data will help in extending the knowledge of the infiltration through different types of construction and will also serve to indicate variations which may be found in the rate of air leakage through different walls of the same type built by different mechanics under different conditions.

All the walls tested at the Laboratory since 1926 and here reported were built by Prof. C. W. Larkin, Head Instructor of Masonry at Carnegie Institute of Technology. In building these walls an attempt was made to follow as nearly as possible the practice of the building industry.

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[b] Research Engineer, Research Laboratory, American Society of Heating and Ventilating Engineers.
[c] Research Assistant, Research Laboratory, American Society of Heating and Ventilating Engineers.

Presented at the 37th Annual Meeting of the American Society of Heating and Ventilating Engineers, Pittsburgh, Pa., January, 1931.

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The method of conducting the tests was identical with that previously used and reported in detail in earlier Laboratory reports [1,2]. The wall to be tested was built as a partition between the pressure chamber and the collecting chamber of the test apparatus (see Fig. 1). Except in cases where edge losses were the subject of investigation, the joint between the apparatus and the wall was carefully sealed with calking compound and then painted with an asphalt base paint so as to limit the leakage measured to that through the 6 x 10 ft. section of wall under test.

The desired wind pressures were produced in the pressure chamber by means

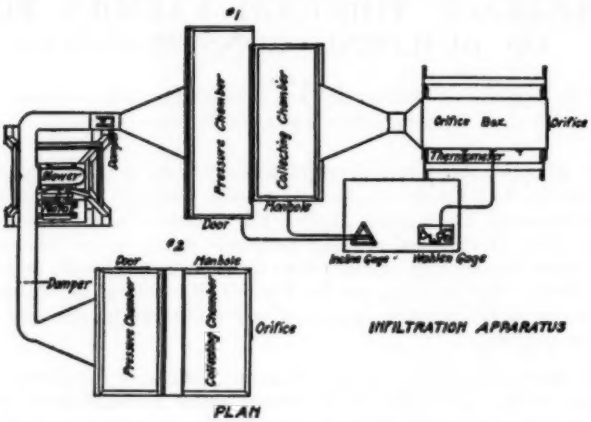

FIG. 1. ARRANGEMENT OF APPARATUS FOR MEASURING LEAKAGE OF AIR THROUGH WALLS

of a blower and the air leaking through the wall into the collecting chamber was measured as it passed out through the calibrated orifice. A Wahlen gage was used to measure the pressure drop through the orifice and a sensitive inclined gage was used to measure the pressure drop through the wall. In analyzing the data, the leakage per square foot of wall or per linear foot of crack was plotted against pressure drop through the wall and the equivalent wind velocity.

Data are given in this paper on the following types of construction:

1. A 13-in. brick wall after plaster had been removed from the wall and the joints repointed.
2. Same as (1) with the application of paint.
3. A frame wall consisting of siding, sheathing, paper, lath and lime plaster.
4. Same as (3) with gypsum plaster.
5. Same as (4) with sizing and paper.
6. Leakage per foot of crack between plaster or stucco and wood frame.

[1] *Air Leakage Through the Openings in Buildings*, by F. C. Houghten and C. C. Schrader, (A. S. H. V. E. TRANSACTIONS, Vol. 30, 1924).

[2] *Infiltration Through Plastered and Unplastered Brick Walls*, by F. C. Houghten and Margaret Ingels. (A. S. H. V. E. TRANSACTIONS, Vol. 33. 1927).

TABLE 1. RESULTS OF TESTS SHOWING AIR INFILTRATION IN CUBIC FEET PER HOUR PER SQUARE FOOT OF WALL SURFACE FOR VARIOUS WIND VELOCITIES

Fig. No.	Curve	Types of Construction	Age in Days	Wind Velocity Miles per Hour					
				5	10	15	20	25	30
2	A	13-in. plain brick wall	22	1.80	4.90	9.35	14.50	20.30	25.50
2	G	Same as 2A, with plaster on brick	183	0.006	0.016	0.031	0.054	0.084	0.121
2	B	Same as 2G, plaster removed, wall repointed	384	1.35	3.25	6.15	10.20	14.95	20.00
2	C	Same as 2B, with one coat paint	387	1.25	2.85	5.10	8.00	11.60	15.65
2	E	Same as 2B, with three coats paint	398	1.05	2.45	4.50	7.00	9.60	12.50
2	F	Same as 2B, with six coats paint	406	0.02	0.06	0.10	0.15	0.20	0.25
4	C	Frame wall siding, paper, sheathing, studs, lath and lime plaster	156	0.25	0.60	1.10	1.90	2.95	4.10
4	D	Frame wall siding, paper, sheathing, studs, lath and gypsum plaster	145	0.25	0.45	0.75	0.95	1.10	1.35
4	E	Same as 4D, with sizing and paper	266	0.10	0.22	0.42	0.60	0.75	0.95
7	B	⅞ in. sheathing, studding, paper, metal lath and stucco		0.12	0.25	0.40	0.50	0.70	0.90
7	D	13 in. brick and tile wall...		1.05	2.90	5.55	8.60	12.15	16.15
				Infiltration in cfh per foot of crack					
8	H	Average, crack between frame and plaster or stucco		0.06	0.135	0.236	0.360	0.50	0.65

7. A stucco wall consisting of sheathing, paper, metal lath, and stucco (without inside plaster finish).

8. A 13-in. brick veneer hollow tile wall.

Figs. 2, 4, 7 and 8, and Table 1 indicate the infiltration rates found for these types of construction.

RESULTS OF TESTS

Brick Wall

The brick wall on which leakage values for both the plastered and plain wall were reported in an earlier Laboratory report was tested after the plaster had been carefully removed and the joints had been repointed. The leakage is given in curve *B*, Fig. 2. This leakage is found to be somewhat lower than that reported for the same plain brick wall before plastering which leakage is shown in curve *A* for comparison. This reduction is probably due to better pointing of the joints after the plaster was removed. One coat of paint was then applied to the inside brick surface with the result shown in curve *C*. Curve *E* gives the results after the application of three coats of paint. It was rather astonishing to those conducting the tests that the paint did not have a greater effect in reducing the leakage, but upon careful inspection of the surface, it was found that the paint did not fill up and seal over the larger cracks and crevices in the wall and it was apparent that the greater portion of the leakage took place through these openings. The wall was then carefully examined and calking material placed in all visible cracks or crevices, followed by an application of paint. This was repeated three times, resulting in leakage indicated by curve *F*. It is apparent that careful calking and painting of brick walls will give a reduction in leakage comparable to the reduction found for plastering. Curve *G* indicates the leakage earlier reported

for the plastered brick wall and is included for the sake of comparison. It should be emphasized, however, that this result was obtained only through a very laborious process of searching out all cracks and crevices and seeing that they were sealed up and covered with paint. It is doubtful if this is a practical procedure.

Curves *H*, *I* and *K*, resulting from tests by Larson and Nelson[3] are also included for purposes of comparison. These are for 13-in. brick walls of the following types: *H*, poor workmanship, lime mortar and porous brick; *I*, good

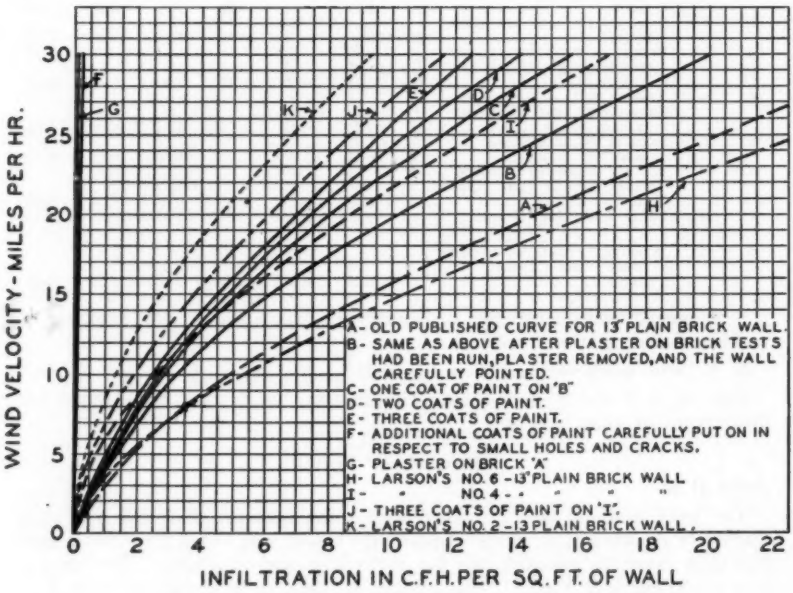
WIND VELOCITY - MILES PER HR.
INFILTRATION IN C.F.H. PER SQ. FT. OF WALL
0
5
10
15
20
25
30
0
2
4
6
8
10
12
14
16
18
20
22
A- OLD PUBLISHED CURVE FOR 13" PLAIN BRICK WALL.
B- SAME AS ABOVE AFTER PLASTER ON BRICK TESTS HAD BEEN RUN, PLASTER REMOVED, AND THE WALL CAREFULLY POINTED.
C- ONE COAT OF PAINT ON "B"
D- TWO COATS OF PAINT.
E- THREE COATS OF PAINT.
F- ADDITIONAL COATS OF PAINT CAREFULLY PUT ON IN RESPECT TO SMALL HOLES AND CRACKS.
G- PLASTER ON BRICK "A"
H- LARSON'S NO. 6 - 13" PLAIN BRICK WALL
I- " NO. 4 - " " "
J- THREE COATS OF PAINT ON "I".
K- LARSON'S NO. 2 - 13 PLAIN BRICK WALL.

FIG. 2. INFILTRATION THROUGH 13-IN. BRICK WALL

workmanship, cement lime mortar, and porous brick; *K*, good workmanship, cement lime mortar and hard brick. Curve *J* indicates the leakage reported for the *I* wall after the application of three coats of paint.

Frame Wall

The details of construction of the frame wall are shown in Fig. 3. The outside construction consisted of ⅞ in. x 8 in. sheathing, building paper, and lap-siding. The sheathing was purchased on the local market for this purpose and was applied immediately with the butt edges firmly against each other so as to allow no visible cracks. After removing the first coat of plaster

[3] *Air Infiltration Through Various Types of Brick Wall Construction*, by G. L. Larson, D. W. Nelson and C. Braatz, A. S. H. V. E. TRANSACTIONS, Vol. 36, 1930.

several months later, it was observed that the sheathing had shrunk showing cracks from ⅛ to ¼ in. in width.

The interior of the wall was lathed and plastered with two coats of rough lime plaster and one lime skim coat. After aging, the frame wall with this plaster showed the leakage indicated by curve *A*, Fig. 4. However, a considerable portion of this leakage took place through the joint between the plaster and the surrounding wood frame as indicated by the infiltration shown by curve *C*, found after sealing the joint. Fig. 5 is a cross section of the construction showing this crack. The frame construction on which the lath and plaster was applied was built in a 1½ x 8-in. board frame placed in the channel iron frame forming part of the apparatus. The joint between the wood frame and the channel frame was carefully calked so that no leakage took place

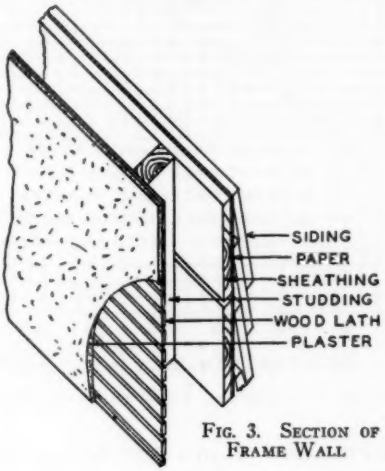

FIG. 3. SECTION OF FRAME WALL

through this joint. The leakage through the frame wall with gypsum plaster before sealing the crack between plaster and wood frame is given by curve *B* and that taking place after this crack was sealed is given by curve *D*. This wall was then sized and papered and again tested giving the leakage indicated by curve *E*.

Stucco Wall

The details of construction of the outside stucco finish is shown in Fig. 6. The ⅞ x 8 in. unmatched sheathing was again purchased on the local market and immediately applied to the studding so as to give as small cracks as possible. Paper and metal netting was applied to the sheating on which two coats of cement plaster and one coat of finish stucco were applied. By using special spacing nails the paper was held firmly against the sheathing and the wire netting was held about ¼ in. away which resulted in the netting becoming completely imbedded in the stucco. By the time the stucco had aged sufficiently

for testing, it was again observed that the sheathing had shrunk so as to leave from ⅛ to ¼ in. cracks between boards.

The stucco wall was built in the same board frame discussed under the plastered wall and curve *A* (Fig. 7) gives the leakage after aging but before the crack between the stucco and the wood frame was sealed up while curve *B*

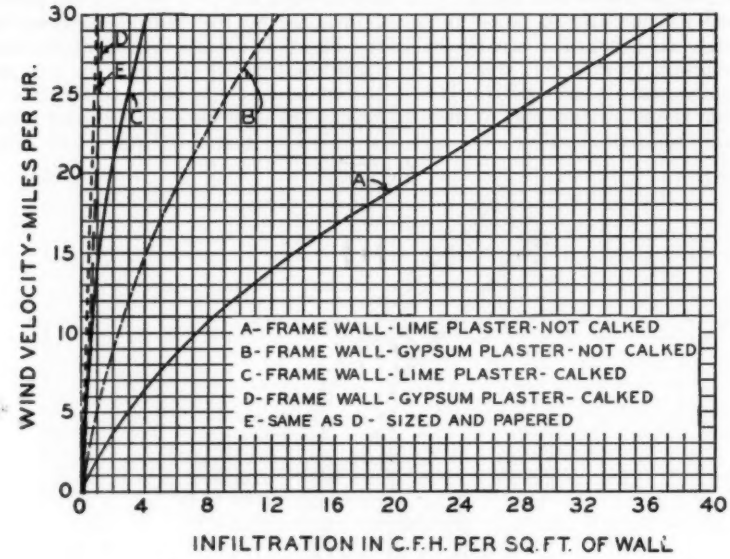

FIG. 4. INFILTRATION THROUGH FRAME WALLS

gives the leakage for the same wall after this crack was sealed. Hence curve *B* represents the leakage per square foot of stucco wall.

LEAKAGE THROUGH CRACKS BETWEEN PLASTER AND WOOD FRAMES

The rates of leakage through the crack between the plaster and the wood frame in the two plastered walls, and between the stucco and the wood frame in the stucco wall, are given in the curves, Fig. 8. In closing this crack for the first plastered wall the top horizontal crack was sealed first giving the reduction per foot of crack indicated by curve *A*. Sealing the two vertical side cracks gave the reduction per foot of crack shown by curve *B* and sealing the bottom horizontal crack gave the reduction shown by curve *C*. The average reduction per foot of crack found by sealing the crack around the entire wall is given by curve *E*.

Sealing the crack around the second plastered wall gave the average reduction per foot of crack shown by curve *F* and sealing the entire crack around the stucco wall gave the reduction shown by curve *G*. Curve *D* gives the average leakage per foot of crack between plaster and wood frame or the average of

curves *E* and *F*. Curve *H* gives the average leakage per foot of crack for the two plastered walls and the stucco wall or the average of curves *E*, *F* and *G*.

It is the opinion of the authors that the leakage shown by the curves obtained by the reduction of the leakage for the walls after sealing the joints is fairly representative of the leakage through the joint between plaster or stucco and the frame work of doors and windows and around the floor line of plastered

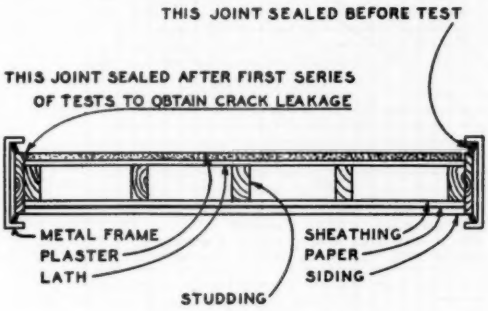

FIG. 5. CROSS-SECTION OF FRAME WALL SHOWING CRACKS

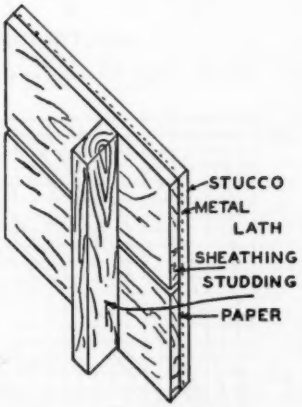

FIG. 6. SECTION OF FRAME STUCCO WALL

walls. A very definite crack always appears at these points during the drying and aging process. It should be pointed out, however, that in most types of construction the finish trim is applied in such a way as to cover this crack. While such trim will always reduce the leakage through the crack to some extent, it rarely eliminates it entirely unless special pains are taken to close the joint between the plaster and trim.

BRICK VENEER HOLLOW TILE WALL

Details of construction of the brick veneer hollow tile wall are shown in Fig. 9. The joints between brick and tile were only partially filled with mortar in

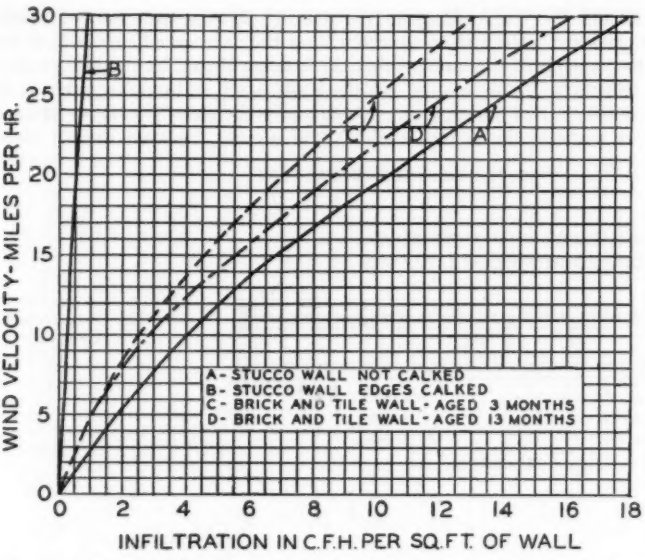
WIND VELOCITY-MILES PER HR.
INFILTRATION IN C.F.H. PER SQ.FT. OF WALL
A- STUCCO WALL NOT CALKED
B- STUCCO WALL EDGES CALKED
C- BRICK AND TILE WALL-AGED 3 MONTHS
D- BRICK AND TILE WALL-AGED 13 MONTHS

FIG. 7. INFILTRATION THROUGH FRAME STUCCO, AND BRICK AND TILE WALLS

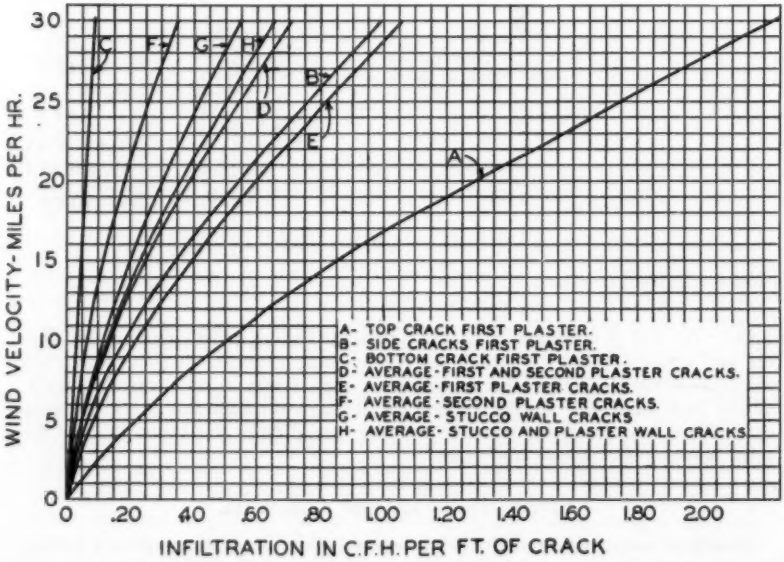
WIND VELOCITY-MILES PER HR.
INFILTRATION IN C.F.H. PER FT. OF CRACK
A- TOP CRACK FIRST PLASTER.
B- SIDE CRACKS FIRST PLASTER.
C- BOTTOM CRACK FIRST PLASTER.
D- AVERAGE-FIRST AND SECOND PLASTER CRACKS.
E- AVERAGE-FIRST PLASTER CRACKS.
F- AVERAGE-SECOND PLASTER CRACKS.
G- AVERAGE-STUCCO WALL CRACKS
H- AVERAGE-STUCCO AND PLASTER WALL CRACKS.

FIG. 8. INFILTRATION THROUGH CRACKS IN FRAME AND STUCCO WALLS

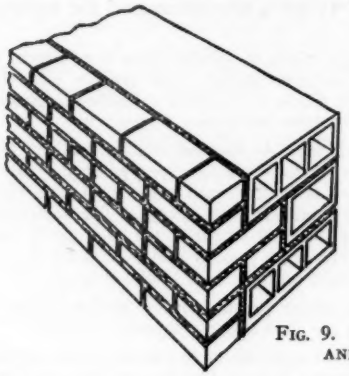

FIG. 9. SECTION OF BRICK AND TILE WALL

a manner similar to common practice. The leakage for this wall three months after construction is given in Fig. 7 by curve *C*, and the leakage after 13-months aging is given by curve *D*.

DISCUSSION

MARGARET INGELS (WRITTEN): I was especially glad to see the information in this paper concerning the air leakage around the edges of plastered walls.

It should be borne in mind that the air leaking through a given wall of a building may not be the product of the cubic feet of air per square foot times the area of the wall. If there is a space within the wall where an air current may be set up, the infiltration may be considerably more than figured from the unit area value. Plaster is practically air tight but a plastered wall may not be, if there is a space between the plaster and the outside portion of the wall which permits a circulation of air within the wall. The air leaking in through the outside portion enters this space and may flow into the interior of the building over baseboards, or escape into the space between ceilings and floors to leak in around interior door frames.

The infiltration of a wall is the sum of 3 factors:

(1) Infiltration due to cracks of windows in the wall. (This factor may be obtained from the data published by the Research Laboratory of the Society.)

(2) Infiltration through walls. This factor may be obtained for several types of walls; also from the data of the Research Laboratory.

(3) Infiltration due to space within the wall which permits air flow between portions of the wall. If there is no air space, or if the air space is sealed, this factor becomes zero.

The curves in Fig. 4 show the effect of the air circulating back of the plaster and finding an outlet around the edges of the plaster. Curve *A* shows the leakage through a frame wall with lime plaster, the crack between the plaster and wood frame not calked. Curve *C* shows the leakage through the same wall with the cracks calked. The difference in these two curves may be called the infiltration around the edges of plaster. Such edges would occur in actual installations at the base boards, window frames and door casings. At 15 miles wind velocity the difference in infiltration is shown as 12.5 cu ft per hour per square foot of wall. When estimating the leakage through a wall of this construction, should the wall area be multiplied by 13.5 cu ft

per hour, or 1.0 cu ft per hour to get a fair estimate of the infiltration through the wall?

In the data for stucco walls the edges of plaster leaks are shown, the top, side and bottom edges each having a different factor. The average cubic feet per hour per foot of crack shown in Fig. 8, Curve *H*, is quite valuable in obtaining the third factor for this type of wall for the total infiltration through it.

The air space, not being sealed, tends to increase the infiltration; the amount of increase depending on the probable length of plaster edges, which may be the perimeter of external wall times the number of floors (each baseboard) plus the sum of the perimeter of all window frames and outside door casings. The additional infiltration due to the plaster edges would be the total length of the edge crack times the cubic feet per hour per foot for the given wind velocity and divided by 2, as the possible maximum crack, exposed to the wind will be ½ of total for home.

The consideration of the space which permits air circulation within the walls is also of importance in determining the heat transfer through the wall.

In buildings with large volume in comparison to external wall surface this third suggested factor is not so important. But, in figuring the heat load for residences it is important because there are usually large amounts of outside wall in comparison to enclosed volumes.

L. B. LENT[4] (WRITTEN): It might be helpful to those who might use these data in design problems to point out that the use of *average* values derived from laboratory experiments may not fit the case in hand and might be a source of error in exact calculations.

To be more specific, conditions found in actual structures may be quite different from those present in the laboratory.

For example, it is stated that in building the brick walls, attempt was made *to follow as nearly as possible the practice of the building industry.* But this practice varies widely in different parts of the country and even in the same city, and the character and performance of brick masonry is known to be greatly influenced by this matter of workmanship. Also aged brick walls seemed to pass more air than new ones.

Again, it was found that, in the frame wall panels, after several months, "the sheathing had shrunk showing cracks from ⅛ to ¼ in. width."

And, that "leakage took place through the joint between the plaster and the surrounding wood frame."

It would therefore appear that, in all of the types of construction examined, conditions which might greatly influence air infiltration in actual structures or structural parts might vary greatly as between similar types and as between different ages of the same type.

In other words, the condition of a brick, frame or stucco wall, a few years after construction might be entirely different (as it affects air leakage) than when newly constructed; or, quite different from the conditions described in these tests.

Hence, a logical conclusion would seem to be that the data given in this paper should be used with caution. The use of *average* values is sometimes dangerous.

In this matter of air leakage through walls, however, the saving feature is that the amount of leakage through the conventional types of walls is practically negligible and may be even desirable as furnishing the necessary ventilation. The usual application of plaster or stucco on the wall surface confines air leakage to the cracks. It is these cracks, either structural cracks around wall openings or undesirable cracks in the walls themselves which really merit our consideration.

[4] Chief Engineer, *Common Brick Manufacturers' Association of America.*

D. W. NELSON (WRITTEN): It is interesting to those of us at the University of Wisconsin who are engaged in infiltration testing, to review the excellent work done in the Research Laboratory of the Society that is reported in this paper. This paper gives valuable additional data on the subject and shows what variations may be expected from different constructions built under different circumstances by different mechanics.

The 13-in. plain brick wall tested after removing the plaster and repointing shows a leakage somewhat less than that of the poorest wall but more than that of the better walls built at Wisconsin. The fact that repointing reduced the leakage seems to bear out the belief that most of the air finds its way through the mortar joints and that this is probably through a hair-line crack at the contact between the brick and the mortar. This crack is caused by a shrinking away of the mortar during the hardening period. A second application of mortar as in repointing, reduces the leakage through this crack.

The application of 3 coats of paint to this brick wall showed about the same reduction in leakage as we found for brick wall No. 4. The application of 2 coats to this wall No. 4 ordinarily applied resulted in a 9 per cent reduction. The third coat was applied with exceptional care in an attempt to fill all pores and hair-line cracks. The result was a total reduction of 28 per cent for 3 coats of paint. This comparatively small reduction for careful application of paint seems to be due to the paint not having sufficient body when dry to clog these hair line cracks.

The application of 3 more coats, 6 in all, to the Pittsburgh wall put on with special care and the use of calking material in all visible cracks reduced the leakage to a negligibly small value just as plastering would do. Due to the expense of this procedure from both the material and labor standpoint such reduction of leakage could not be obtained at a reasonable building cost.

Probably the application of hot asphalt to the brick wall would be effective in the reduction of air infiltration as well as in the prevention of moisture entrance for which reason alone, it is usually justified. The application of such a coat to the brick surface of a furred wall is to be recommended due to the loss of effectiveness of the plaster by leakage at the edges of the plaster sheet at window and door openings and at the baseboard.

The tests at Wisconsin on brick wall No. 7, which was equipped with a standard baseboard applied to the usual furred and plastered construction, showed that the efficiency of the plaster sheet in stopping air leakage was only about 50 per cent due to the crack at the baseboard. The crack at the baseboard allowed the passage of 12 cu ft per hour per foot of length and if it had offered no resistance would have passed 22.5 cu ft per hour per foot of length at 15 mph.

This leakage is much larger than that found at the Pittsburgh Laboratory and reported in this paper for the crack around the plaster and stucco sheets. The average of all was 0.25 cu ft per hour per foot of crack and the maximum was 0.85 cu ft per hour per foot of crack at 15 mph.

Should the baseboard be made exceptionally effective by running the plaster entirely to the floor or by using calking material between the baseboard and the plaster, there would still be the chance for air to enter into the floor and ceiling construction from the wall space and find its way into the room through openings such as around the steam pipes. There is also the chance for air to reach window frames in furred masonry and in frame walls. At this location, entrance to the room is not effectively resisted. In fact, the pulley holes and sash weight door cracks offer easy entrance. In our opinion the effectiveness of the plaster in furred masonry and in frame walls should not be over 50 per cent.

Because of the opportunity for air leakage at the edges of the plaster sheet, especially at the baseboard and window openings, building paper must be largely relied

upon for infiltration effectiveness in frame construction. The tests at Wisconsin reported to the Society in June, 1930, showed that all of the half dozen walls of sheathing, paper and siding or shingles and without plaster had a negligible leakage. The air resistance is due to the paper which was effectively clamped between the 2 thicknesses of boards.

The leakage through the finished frame walls was less than ½ cu ft per square foot per hour at 15 mph. The values secured in Pittsburgh as shown in Fig. 4 are about 1.10 and 0.75 cu ft. These leakages are extremely small and in either case would be neglected in ordinary heat loss calculations. This difference, though negligible, may be due to a difference in method of calking. At Wisconsin, a recess was made in the 2 x 4s against the 4 x 8 wood members of the frame into which the walls were built. This was filled with calking compound. In addition a space of ⅛ to ¼ in. was left on the 4 sides of the outer wall against the frame and this space was filled with calking compound against the siding, paper and sheathing down to the 2 x 4s, so that air could not enter at the ends of the test panel.

Fig. 5 of this paper does not show calking at these locations and the slight additional leakage secured may be accounted for by this condition. It is a question which method of calking comes nearest to representing building conditions. In preliminary tests on frame walls *2B* and *3B* at Wisconsin, one of these end joints of siding, paper and sheathing was left uncalked and the leakage resulting was 1.70 cu ft as compared to 0.30 cu ft when sealed. This was considered to represent the condition at the corner of a house or against an opening. This crack leakage amounted to 0.20 cu ft per foot of crack per hour, which is close to that of the average plaster and stucco cracks found at the Pittsburgh laboratory.

The 13-in. brick and tile wall reported in this paper had an air leakage of about 5.0 cu ft per hour per square foot of wall at 15 mph arrived at by taking an average from the 2 curves shown in Fig. 7. An 8-in. hollow tile wall tested at Wisconsin had a leakage of 9.8 cu ft per square foot per hour at 15 mph. The tile itself then seemingly allows the passage of twice the air that the combined brick and tile wall does. There seems to be a great variation in workmanship on tile walls and some would have a much greater leakage than the foregoing figures would indicate.

The test on this brick and tile wall at the end of 13 months showed something over 20 per cent more leakage than did the test at 3 months. There was no indication in the brick wall tests at Wisconsin that there was any change in leakage attributable to aging. The tests were made at 5 months and then 2 months later. It would be interesting in the near future to again test these brick walls and the tile wall to look for an aging effect.

No. 889

CAPACITY OF RETURN RISERS FOR STEAM AND VAPOR HEATING SYSTEMS

By F. C. Houghten [1] *(MEMBER)*, and Carl Gutberlet [2] *(NON-MEMBER)*, Pittsburgh, Pa.

THIS paper marks the completion of the general investigation of the capacity of pipe for carrying steam, condensate, and air in various parts of steam heating systems, as outlined by the Technical Advisory Committee on Pipe Sizes for Heating Systems, and as carried on by the Research Laboratory in cooperation with the *Carnegie Institute of Technology*. Publication of the results of various phases of the study have appeared in the Transactions [3] of the Society since 1922, and have formed the basis for the pipe size tables contained in The Guide. This report deals with the capacity of return risers. The investigation was carried on in conjunction with the study of return mains, a report [4] of which was published recently.

Test Set-up and Operation

The test set-up, Fig. 1, varied only in minor details from that used in the study of return mains.[4] Steam from a high pressure line A was supplied to an 8 in. dripped main B, by means of suitable pressure reducing valves and hand-controlled throttling valves. From the main B, the steam was delivered to six unit heaters, C_1 to C_6 varying in size from 150 to 2,200 equivalent sq ft of direct cast iron radiation. Any or all of the units could be turned on. The largest unit was provided with fan speed control. With this arrangement condensation could be supplied at any rate desired through the return main DE to the head of the riser, E. In order to give complete control over the rate of air carried by the return riser, provision was made for admitting air through the orifice flow meter V to the supply main B.

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[1] Director, Research Laboratory, A. S. H. V. E.
[2] Research Engineer, Research Laboratory, A. S. H. V. E.
[3] A. S. H. V. E. Transactions, 1922 to date.
[4] *Capacity of Dry Return Mains for Steam and Vapor Heating Systems*—by F. C. Houghten and Carl Gutberlet, A. S. H. V. E. Transactions, Vol. 36, 1930.

Presented at the 37th Annual Meeting of the American Society of Heating and Ventilating Engineers, Pittsburgh, Pa., January, 1931.

The base of the riser F could be connected to the receiving bucket L and the vacuum pump through 5 loops containing 540 ft of 1 in. dry return main, by closing the connection through F, G, H, and K. By closing the connection through F, I, J, and K, the return main could be eliminated. With either

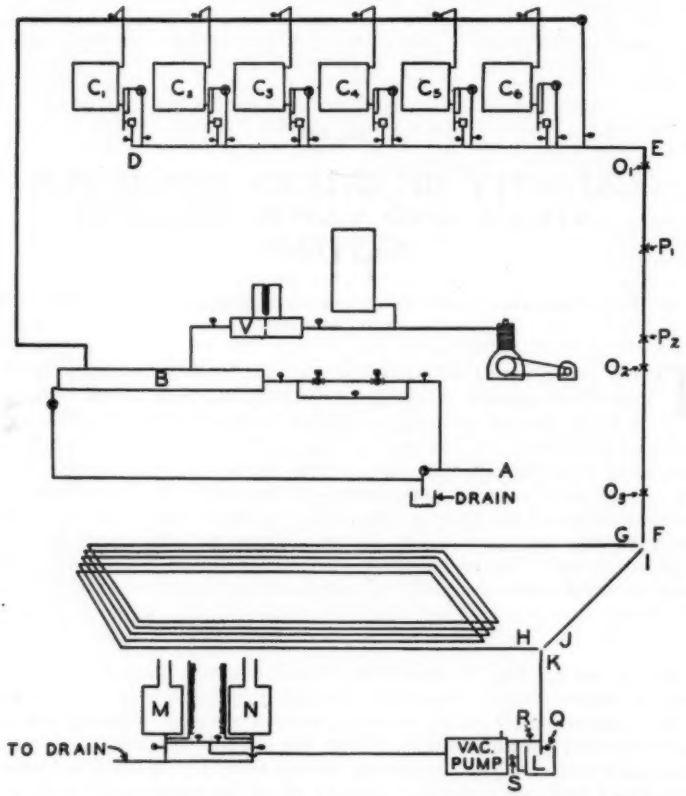

FIG. 1. LABORATORY SET UP FOR STUDYING CAPACITIES OF RETURN RISERS

connection the system could be operated by gravity or by vacuum pump return. By opening valve Q and closing valve R, the system returned by gravity to the receiving bucket at atmospheric pressure, from which the condensation was pumped through S into the measuring tanks M or N. By closing Q and S and opening R, the system operated as a vacuum pump return, the pump delivering the condensate directly to the measuring tanks.

One-inch and ¾ in. risers were studied. Provision was made for observing the pressure at two points P_1 and P_2 along the riser in addition to points

O_1, O_2, and O_3, by inserting the connection and manometer shown in Fig. 2. The large pipe connection prevented water from sealing across the pipe by capillary attraction, thus giving an erroneous reading. At each of the three points O_1, O_2, and O_3, a 28 sq ft radiator was drained into the riser. Fig. 3 shows the connection used for these radiators which was used primarily to indicate whether or not the riser was overloaded, but which also served to give three additional points at which pressures in the riser could be observed. When a radiator was not in use it was shut off from the return riser by closing the gate valve between the trap and the riser.

In operating the system, the rate of condensation and air supply to the return riser was adjusted and maintained constant for a sufficient length of time to insure uniform operation and uniform temperature throughout the system. When all conditions were constant, the rate of condensation and the rate of air flow through the riser were determined. The condensate was measured by a graduated gage glass on the measuring tanks, and the air, by an orifice flow meter in the air supply line. The pressures at the points P_1 and P_2, and also those in the radiators, after the valves between the radiators and the riser were opened, were observed. If the pressure in the radiator before turning on the steam had been equal to or below atmospheric pressure, the steam supply valve was opened sufficiently to give a pressure of 1 oz. in the radiator as indicated by the manometer.

If the pressure in the radiator before turning on the steam was above atmospheric pressure, then the steam supply valve was opened sufficiently to give a manometer reading 1 oz. above this value. The time for completely heating the radiator was observed, and if more than 20 minutes were required the riser was considered to be overloaded and the test was not used.

Tests were made at condensation rates of 250, 500, 900, 1200, 1500, and 2000 lb per hour on the 1-in riser and rates of 150, 300, 500, 900, and 1200 lb per hour on the ¾-in. risers. For each load studied the pressure drop along the riser was determined for various rates of carrying air up to the maximum which could be handled without overloading the riser, producing noise or requiring an excessive pressure drop.

EXPERIMENTAL RESULTS

The pressures along the riser for several rates of handling condensation and air are plotted in Fig. 4. The pressure drop throughout the length of the riser in ounces per 100 ft length of run was calculated from curves similar to these in order to arrive at the pressure drop, condensation and air flow relationships desired.

The pressure drops in the riser for the various conditions studied as calculated from curves similar to those in Fig. 4 are plotted in Figs. 5 and 6 against cubic feet of air per hour handled by the 1-in. and ¾-in. risers, respectively.

It is of interest to note that for comparatively high rates of handling condensation with correspondingly low rates of handling air, negative pressure drops are observed, that is, with an increasing distance along the riser in the direction of flow (or from the top towards the bottom of the riser) the pressure increased rather than decreased. This is as would be expected upon careful consideration, for under these conditions acceleration of the descending

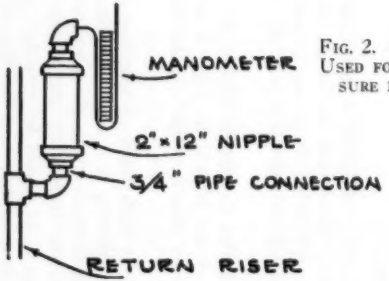

FIG. 2. (LEFT) CONNECTION USED FOR MEASURING PRESSURE IN RETURN RISERS

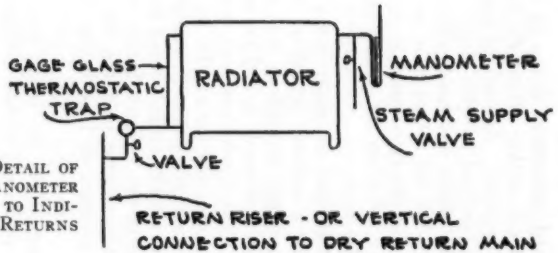

FIG. 3. (RIGHT) DETAIL OF RADIATOR AND MANOMETER CONNECTIONS USED TO INDICATE OVERLOADED RETURNS

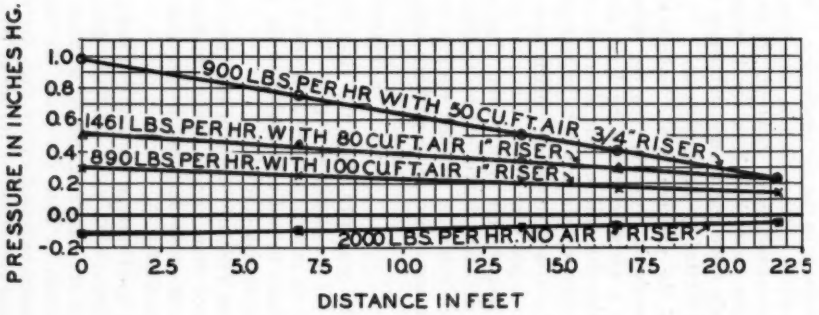

FIG. 4. PRESSURE DROP THROUGH THE RETURN RISERS FOR VARIOUS AIR AND CONDENSATION LOADS

water in the riser produces an aspirating effect tending to draw the air downward so as to produce a partial vacuum at the top of the riser.

For comparison, similar curves, *A* and *B*, Fig. 5, are shown giving the relation between pressure drop and cubic feet of air handled for 500 and 1500 lb condensation loads on a 1 in. horizontal dry return main pitched 1 in. in 16 ft. These curves are taken from an earlier laboratory report [4] of a study

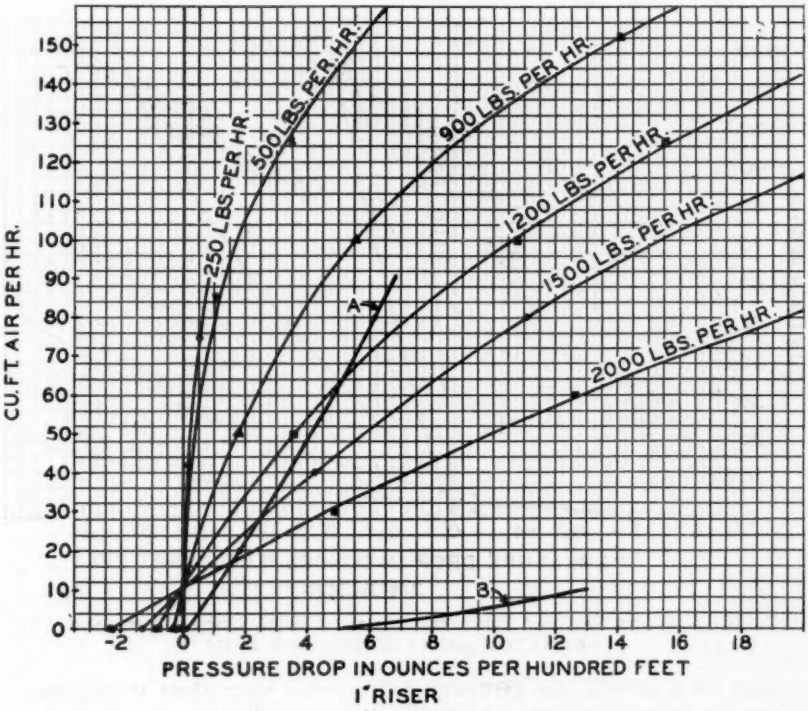

FIG. 5. RELATION BETWEEN PRESSURE DROP IN A 1-IN. RETURN RISER AND AIR CARRIED FOR VARIOUS CONDENSATION RATES

of dry return mains. It will be observed that the pressure drop for handling air and condensate at any rate is very much lower for the return riser than it is for the dry return main.

The pressure drop in ounces per 100 ft length of run along the 1-in. and ¾-in. risers is plotted in Figs. 7 and 8, respectively, against condensation load in pounds per hour, and in equivalent square feet of radiation for various rates of handling air ranging up to 150 cu ft per hour. It is again of interest to note the negative pressure drop found for certain rates of handling condensation and air. For the highest loads given for the ¾-in. riser, the *no air*

curve tends to flatten out with increased load. This tendency would, no doubt, also be found for the *no air* curve on the 1-in. riser if the curve were carried out to higher loads. These curves must necessarily again curve upward and intersect the *zero pressure drop* line for some higher load just as the 5 cu ft curve for the ¾-in. riser after descending below the *zero pressure drop* line again ascends and recrosses the *zero* axis. The particular load at which the *no air* curve would again intersect the *zero pressure drop* line may be calcu-

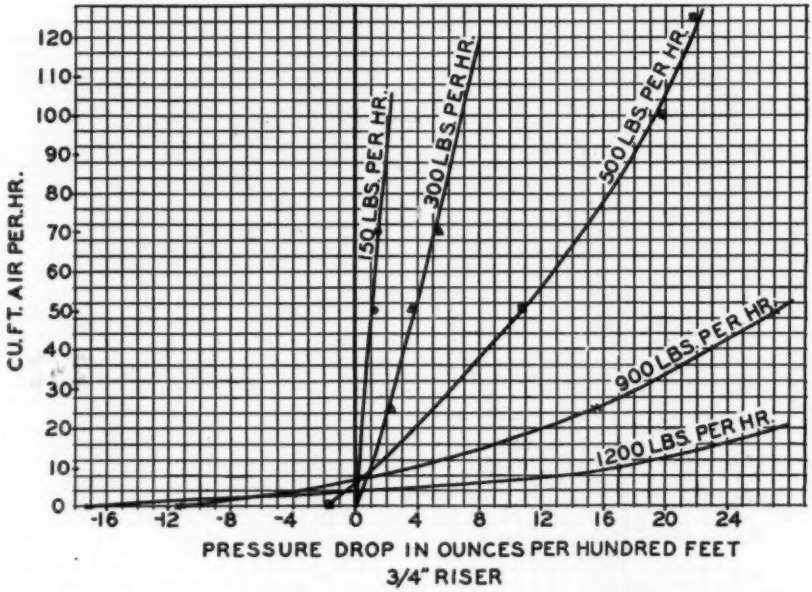

FIG. 6. RELATION BETWEEN PRESSURE DROP IN A ¾-IN. RETURN RISER AND AIR CARRIED FOR VARIOUS CONDENSATION RATES

lated for a vertical pipe filled with and carrying water alone at maximum capacity for gravity flow by applying the Darcy formula:

$$Q = ACF\sqrt{RS},$$

in which the various factors have the following significance and values for the 1-in. pipe:

Q = pounds of water per hour flowing through the pipe
A = area of the pipe
R = the mean hydraulic radius
S = Sine of the slope of the pipe = 1
C = a constant depending on the pipe
F = constant to change cubic foot of water per second to pounds per hour = 216,000

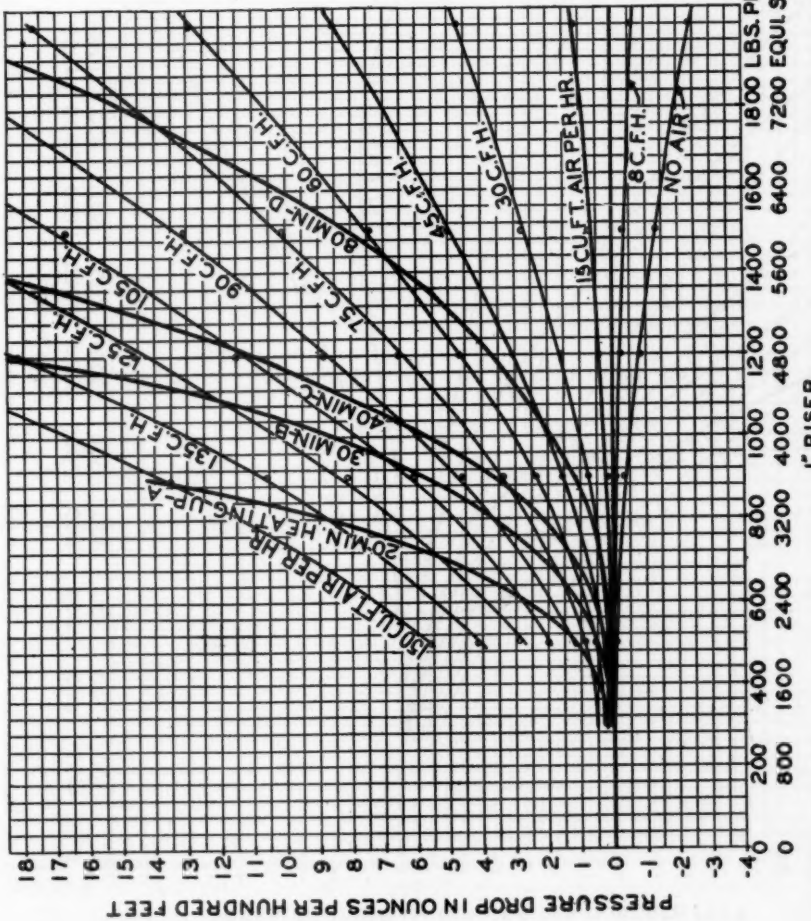

FIG. 7. RELATION BETWEEN PRESSURE DROP AND CONDENSATION LOAD FOR VARIOUS RATES OF AIR ELIMINATION THROUGH A 1-IN. RETURN RISER

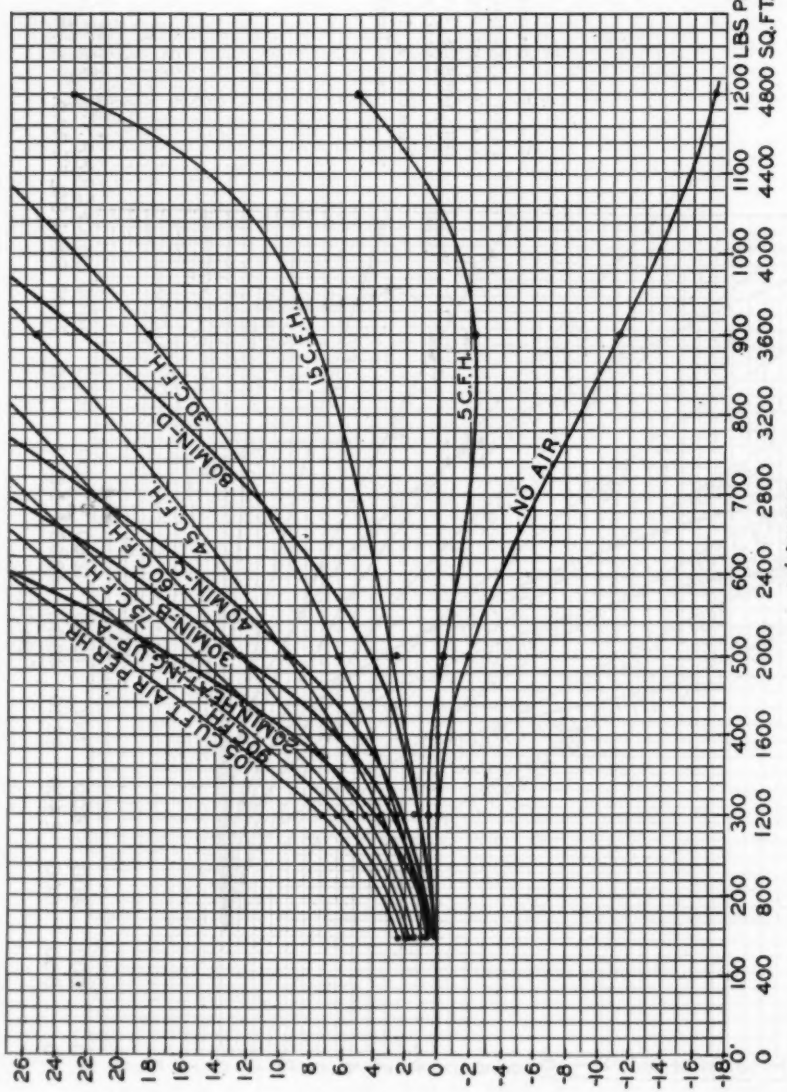

FIG. 8. RELATION BETWEEN PRESSURE DROP AND CONDENSATION LOAD FOR VARIOUS RATES OF AIR ELIMINATION THROUGH A ¾-IN. RETURN RISER

Kent [5] gives $AC\sqrt{R} = 0.06334$ for a 1-in. pipe and 0.02855 for a ¾-in. pipe. Therefore:

$$Q = 0.06334 \times 216{,}000 \times 1 = 13{,}682 \text{ lb per hour for a 1-in. pipe}$$
$$Q = 0.02855 \times 216{,}000 \times 1 = 6{,}167 \text{ lb per hour for a ¾-in. pipe.}$$

For the 1-in. vertical pipe filled with and carrying water alone at full capacity, due to gravity, the *no air* curve should again intersect the *zero pressure drop* line at a load of 13,682 lb per hour of water. Likewise the *no air* curve for the ¾-in. pipe should intersect the *zero pressure drop* line at a load of 6,167 lb of water per hour.

It should be observed that the air loads indicated for the various curves in Figs. 4 to 8 represent the rates at which air was added to the steam. There was always in addition to this amount of air some non-condensable gases and probably also some water vapor so that the *no air* curve as plotted in Figs. 7 and 8 are really curves for some minimum amount of air, gas, and non-condensed vapors, and these curves, no doubt, actually intersect the *zero pressure drop* line at loads considerably below the theoretical value given by the Darcy formula.

In order to observe the quantity of air, non-condensable gases and vapors carried in the condensate, besides the quantity of air added to the system, a glass tube of the same internal diameter was placed in the bottom of the return risers and the system operated for a short time under different conditions. For condensation loads up to about 1200 lb per hour for the 1-in. riser, the descending water rarely filled the pipe, but descended in torrents down the sides and in a rain down the center. As the quantity of air handled increased, the descending water and air became more agitated. When no air was added, the flow was similar but with comparatively little agitation. For condensation loads above approximately 1200 lb per hour, slugs of water appeared, but even when no air was added these slugs were always interspersed with slugs of air and water spray.

The curves showing the relation of pressure drop to load for various rates of air handled as plotted in Figs. 7 and 8 give the data necessary for choosing the size of pipe required for any desired pressure drop and for any rate of air and condensation handled within the limits studied. However, before these curves may be used in practice it is necessary to arrive at the rate at which a riser in a given system will be required to handle air. This will depend upon a great many factors, including the volumetric capacity of the radiation and piping used in a given system and the rate at which it is desired to eliminate the enclosed air in heating up.

After all radiation in the system is heated up and the air eliminated, the rate at which air will be handled will be extremely low as shown by the tendency of the curves. The capacity for carrying condensation with a given pressure drop under such conditions is extremely high. It is obvious that the size of pipe required for any return riser must be based not upon its capacity to carry condensation when the system is entirely heated up or when little air or non-condensable vapors are being eliminated, but it must rather be based upon the rate of air elimination during the heating-up period, which will depend in a large measure upon the type of radiation and rate of heating up.

[5] Kent's Handbook, 1923, p. 754.

In an earlier laboratory report,[4] results were given of the average volumetric capacity found for a large number of sizes and makes of tubular cast iron radiators, which average was given as 0.0142 cu ft of air per square foot of radiation.

If it is assumed that the maximum rate of air elimination from a system while the return piping is handling both air and condensation at the maximum rate may be based upon the time of heating up the entire system and the elimination of a volume of air equal to the capacity of all the radiation served, it may then be possible to arrive at pressure drop, condensation relationships for various rates of heating up. Curves *A* to *D*, Figs. 7 and 8, give the pressure drop condensation relationships for rates of heating up ranging from 20 to 80 min. for systems in which all steam is condensed in tubular cast iron radiation having an average volumetric capacity of 0.0142 cu ft per square foot of radiation.

If air is eliminated in 20 minutes from a system in which all steam is condensed in tubular cast iron radiators having a volumetric capacity of 0.0142 cu ft per rated square foot, then the system must handle air at the rate of $3 \times 0.0142 = 0.0426$ cu ft of air per hour per square foot of radiation or 0.1704 cu ft of air per pound of condensation handled. The rates of handling air per pound of condensation given in curves *B*, *C*, and *D* were calculated similarly.

In Fig. 9 the limits in rates of handling air and condensation for satisfactory operation are given by curves *E* and *F* for the 1-in. and ¾-in. risers respectively. For determination of these limitations the riser was said to operate satisfactorily when the radiators O_1, O_2, and O_3 (Fig. 1) returning into it become entirely heated in 12 min without noise.

In all of the tests resulting in the data used in Figs. 4 to 9 the system was operated either by gravity or vacuum pump return as required to give satisfactory operation. If the system operated satisfactorily by gravity return it was also found to operate satisfactorily with vacuum pump return and with substantially the same pressure drop load relationships. When the vacuum pump was applied to a condition under which the system would operate satisfactorily by gravity the radiators heated up more quickly and, provided the vacuum pulled by the pump was not great enough to produce an excessive vacuum (not over 1 in. of water) at the top of the riser, the pressure drop relationship for the same load condition was found to remain the same or to increase by a practically immeasurable amount. However, when the vacuum at the pump was increased so as to produce a greater vacuum at the top of the riser, the pressure drop for the same load condition increased. This increase as pointed out in an earlier laboratory report on dry return mains was probably the result of reevaporation of condensate upon passing through the trap when a large pressure drop through it existed.

Most of the tests resulting in the data presented were made with direct connection between the base of the riser and the receiving bucket or the vacuum pump. Especially for gravity returns this connection was made large so that there would be substantially no pressure drop between the base of the riser and the receiving bucket. In other words, the pressure at the base of the riser was substantially atmospheric pressure. Some tests were made, however, on all conditions reported in which the condensation from the riser re-

turned through the 540 ft dry return main. Excepting for very small loads, the insertion of the return main between the base of the riser and the receiving bucket gave unsatisfactory operation of the riser when operating by gravity. However, if the vacuum pump was applied so as to give atmospheric pressure

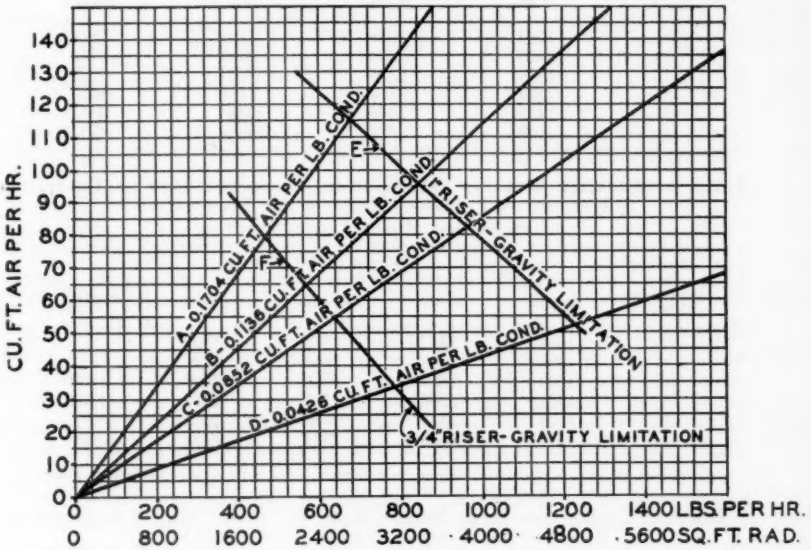

FIG. 9. LIMITATION OF AIR AND CONDENSATION LOADS FOR GRAVITY OPERATION OF 1-IN. AND ¾-IN. RETURN RISERS

at the base of the riser *F*, the same pressure drop load relationships were found for the riser.

APPLICATION OF THE DATA IN SIZING RETURN RISERS

The relationships given in the curves, Figs. 4 to 9, may be used for developing tables giving capacities of 1 in. and ¾ in. return risers. The acceptance of such tabular values must, however, be based upon certain assumptions, including the allowable pressure drop, the rate of heating up or air elimination and any factor of safety desired for covering other contingent factors. These assumptions cannot be determined by laboratory test but must be based upon experience gained through practice and judgment.

Table 1 gives the capacity of a 1-in. return riser in equivalent square feet of radiation for various pressure drops and various rates of heating up a system in which all steam is condensed in tubular cast iron radiators having a volumetric capacity of 0.0142 cu ft per square foot of radiation.

Table 2 gives capacities 20 per cent lower than those given for the same conditions in Table 1. This 20 per cent reduction in capacity may be considered a factor of safety for variation in internal diameter and internal

TABLE 1. CAPACITY OF 1 IN. RETURN RISER IN SQUARE FEET OF EQUIVALENT RADIATION FOR VARIOUS RATES OF AIR ELIMINATION

Experimental Results

Pressure Drop Ounces Per 100 Ft	Period of Air Elimination			
	20 Min	30 Min	40 Min	80 Min
¼	1360	1640	1940	2560
½	1620	2000	2300	2960
1	1936	2336	2704	3472
2	2240	2760	3136	4032
3	2472	3040	3440	4512
4	2648	3272	3680	4912

TABLE 2. CAPACITY OF 1 IN. RETURN RISER IN SQUARE FEET OF EQUIVALENT RADIATION FOR VARIOUS RATES OF AIR ELIMINATION

Experimental Results Less 20 Per Cent

Pressure Drop Ounces Per 100 Ft	Period of Air Elimination				
	20 Min	30 Min	40 Min	80 Min	From Guide 1930 [a]
¼	1088	1312	1552	2048	
½	1296	1600	1840	2368	
1	1549	1869	2163	2778	1200
2	1792	2208	2509	3226	1704
3	1978	2432	2752	3610	
4	2118	2618	2944	3930	2400

[a] Capacities of return risers for vacuum installations. No rate of air elimination given.

TABLE 3. CAPACITY OF ¾ IN. RETURN RISER IN SQUARE FEET OF EQUIVALENT RADIATION FOR VARIOUS RATES OF AIR ELIMINATION

Experimental Results

Pressure Drop Ounces Per 100 Ft	Period of Air Elimination			
	20 Min	30 Min	40 Min	80 Min
½	600	680	720	860
1	764	820	892	1136
2	964	1056	1152	1496
3	1108	1228	1360	1776
4	1208	1368	1500	2116

TABLE 4. CAPACITY OF ¾ IN. RETURN RISER IN SQUARE FEET OF EQUIVALENT RADIATION FOR VARIOUS RATES OF AIR ELIMINATION

Experimental Results Less 20 Per Cent

Pressure Drop Ounces Per 100 Ft	Period of Air Elimination				
	20 Min	30 Min	40 Min	80 Min	From Guide 1930
½	480	544	576	688	
1	611	656	714	909	700
2	771	845	922	1197	994
3	886	982	1088	1421	
4	966	1094	1120	1693	1400

TABLE 5. MAXIMUM RATE OF AIR ELIMINATION FROM TUBULAR CAST IRON RADIATION FOR VARIOUS CONDENSATION LOADS AND CORRESPONDING HEATING UP PERIOD FOR 1 IN. AND ¾ IN. GRAVITY RETURN RISERS

Experimental Results

Capacity Sq Ft Equivalent Radiation	Maximum Rate of Air Elimination			
	1 Inch Riser		¾ Inch Riser	
	Time, Min [a]	Cu Ft Air/Hr	Time, Min [a]	Cu Ft Air/Hr
1600			15	89.5
2000			23	74.5
2400	16	124	34	60.0
2800	20	112.5	50	45.5
3200	27	100.5	88	31.0
3600	34	89.5		
4000	43	78.5		
4400	56	67.0		
4800	74	55.5		

[a] Time necessary for eliminating air from a system in which all steam is condensed in tubular cast iron radiation having a volumetric capacity of 0.0142 cu ft per equivalent square foot of radiation.

smoothness of pipe, constrictions due to improper cutting, reaming, and the use of dope, and other contingent factors. The acceptance of a factor of 20 per cent is in conformity with the earlier practice of the committee which developed the pipe size tables for the supply side of steam heating systems contained in THE GUIDE, 1930, and is not the result of this laboratory investigation. Tables 3 and 4 give similar data to Tables 1 and 2 for a ¾-in. return riser.

Table 5 is based upon Fig. 9 and gives minimum time in which various amounts of tubular cast iron radiation can be heated up when returning through a 1-in. or a ¾-in. riser, and also the accompanying maximum rate of air elimination. These data are given for both the 1-in. and ¾-in. risers. Table 6 is derived from Table 5 by the application of a 20 per cent factor of safety.

TABLE 6. MAXIMUM RATE OF AIR ELIMINATION FROM TUBULAR CAST IRON RADIATION FOR VARIOUS CONDENSATION LOADS AND CORRESPONDING HEATING UP PERIOD FOR 1 IN. AND ¾ IN. GRAVITY RETURN RISERS

Experimental Results Less 20 Per Cent

Capacity Sq Ft Equivalent Radiation	Maximum Rate of Air Elimination			
	1 Inch Riser		¾ Inch Riser	
	Time, Min [a]	Cu Ft Air/Hr	Time, Min [a]	Cu Ft Air/Hr
1280			15	89.5
1600			23	74.5
1920	16	124	34	60.0
2240	20	112.5	50	45.5
2560	27	100.5	88	31.0
2880	34	89.5		
3200	43	78.5		
3520	56	67.0		
3840	74	55.5		

[a] Time necessary for eliminating air from a system in which all steam is condensed in tubular cast iron radiation having a volumetric capacity of 0.0142 cu ft per equivalent square foot of radiation.

XUM

Tables 2 and 4 give, for the sake of comparison, capacities for return risers for various pressure drops as used in the pipe size tables in THE GUIDE, 1930. No required time for heating up is given in connection with tables in THE GUIDE. It is of interest to note that for a 1-ounce pressure drop, which is usually considered best practice, the values in THE GUIDE are considerably below the values resulting from this investigation and the application of a 20 per cent factor of safety for a 20-minute heating-up period. However, for higher pressure drops this difference disappears and finally reverses so that for a 4-in. pressure drop THE GUIDE values are higher, although not enough so that the 20 per cent factor of safety would allow the riser to operate satisfactorily in the absence of any contingent factors tending to decrease is capacity. In the case of the ¾-in. riser (Fig. 4) all GUIDE values show higher capacities in comparison to those resulting from this investigation, and the application of the 20 per cent factor of safety for a 20-min. heating-up period. For a 1-oz. pressure drop, the values in THE GUIDE correspond to a heating-up period of approximately 40 min., whereas for a 4-oz pressure drop the values in THE GUIDE correspond to a longer heating-up period.

It is not the purpose of this investigation to establish the desirable heating-up period on which to base pipe size capacities. However, it may be of interest to point out that a 20-min. heating-up period is probably much shorter than desirable for most buildings for the reason that in most instances it is not necessary or desirable to heat up all radiation from the cold condition in this period of time.

SUMMARY AND CONCLUSIONS

1. As a result of this investigation, curves in Figs. 7 and 8 give the relation between condensation carried by the return riser, pressure drop through the riser, and air carried by it on which pipe size capacities may be based.

2. Capacities of return risers are given in tabular form based upon experimental values found in the laboratory and also upon the application of a factor of safety of 20 per cent to these values.

DISCUSSION

T. M. DUGAN (WRITTEN): This work as explained represents an answer to what I understand is the last phase of an important problem that engaged the attention of the Society's Committee on Research for over 10 years. If the members and industry at large will make use of the information contained therein, it will undoubtedly mean considerable saving to the consumer.

Unfortunately, the latest reports and data available are not always accepted for their true worth, or else probably they are taken with a grain of salt.

With findings of the Research Laboratory available in regard to such a simple thing as painting radiators, it is surprising that many contractors still use gold or aluminum bronze.

A brief double check of a job and an application thereto of the curves and data would quickly prove the merit of the report.

I had occasion to make a careful study of accepted plans for an $18,000.00 contract submitted by reputable engineers. An experienced contractor handled this job. The returns seemed to be oversized. An analysis of one section or wing proved typical

of the whole job. This particular section was without basement and shows a return line 82 ft in length graduated in size from 1¼ to 2½ in. exposed on the baseboard.

The return served as a drip to a 105 ft 4 in. steam main and 7 radiators on the basement wall having a combined heating load or capacity of 360 sq ft.

The system as designed was a gravity vapor job with a common air eliminator in boiler room.

I took exception to the large return exposed on the baseboard not for economic reasons, for the contract was already let, but for the looks of the job. By reference to THE GUIDE 1927, and after I assumed complete responsibility for the successful operation of that portion of the system, it was agreed that the final size of return would be 1¼ in. and other sections in proportion. An air eliminator was placed at a point near the end of the main in a small storage room, thus reducing the line to ¼ the original size, showing corresponding saving of pipe fittings and labor and needless to say a smooth working system.

To apply the value of Director Houghten's paper to any job would be interesting. Right here we see 105 ft of main and 360 ft of cast iron tubular radiation to be drained of approximately 14 cu ft of air and condensation in a heating up period of 20 min. Try this with your pencil and note just what the value of the paper is.

Briefly, another section of the building was heated by a 1,050 ft bank of fan operated vento. This return was sized on the original plans as 3½ in. but installed as a 1½-in. wet return. The evident saving by reduction in pipe sizes was so apparent that the contractor without solicitation rebated $225.

If these curves and figures are translated into practical pipe sizes suggested by the authors of the paper so that the fitter can interpret them without the necessity of computation, then this particular piece of research will have been amply worthwhile.

R. C. MORGAN (WRITTEN): The results shown are most interesting and indicate that further tests along the same lines showing capacities of return mains would be of tremendous value and probably lead to the revision of the tables in THE GUIDE on the capacities of return piping in general.

It seems that the sizes as stated in THE GUIDE for the return piping, while based on certain definite formula, need revision.

H. M. HART (WRITTEN): This paper is an excellent contribution to the science of heating. It gives much needed information on the proper sizing of piping and it stresses the importance of time allowance for filling the system with steam.

This heating-up period is a complex problem, because it involves so many variables—the type of piping system, the kind of heating units, the source of steam supply and the building temperature at the time of starting the system.

The laws of economics demand the elimination of abnormal peak loads in so far as practicable, and it might be well for heating engineers to bear this in mind when designing a heating system.

Section V of the Code of Minimum Requirements recommends certain percentages to be added to design load to take care of this starting load. These percentages are on a sliding scale starting at 65 per cent for 100,000 Btu and gradually reducing to 40 per cent for design loads of 1,800,000 Btu and over. This would indicate a time allowance for the heating-up period of from 25 to 33 min to which must be added the time necessary to get the boiler up to its maximum output capacity. If we were to allow but 20 min for the heating-up period it would be necessary to add 85 per cent to the design load to arrive at the actual maximum demand on the boiler. This would mean: (1) boiler ⅓ larger than would be required for a 33-min heating-up period, (2) a corresponding increase in the size of the vacuum pump and motor, as well as the oil burner or mechanical stoker, (3) an increase in the size of the piping, and (4) a normal demand on the system of 27 per cent of the maximum instead of 35.7 per cent.

Inasmuch as this maximum demand is only required for this 20 or 33 min, not to exceed once each day, it is up to the engineer to decide whether or not it would be worth the extra installation cost which would result in a saving of not over 13 min a day of a fireman's time.

We should carefully consider the recommendations of our Boiler Selection Committee, given in Section V of the Code of Minimum Requirements, with a view to recommending similar time allowance for the piping systems, or some other time allowances which might be more suitable. It seems that 40 min would be adequate for the average installation.

PRESIDENT HARDING: Mr. Hart has raised an interesting question and I am sure the continuing committee for boiler selection and rating will take the matters suggested under consideration.

I am not so sure that all the members of this organization thoroughly understand what a heating-up factor is. It has been discussed a number of times before the Society but in my travels around I find that there are many who have quite a poor conception of what a heating-up factor really means. It does take time to heat up iron; you don't do it instantaneously and it is a matter of judgment as to how much time we ought to allow to heat up a steam heating system. The Code of Minimum Requirements states various percentages from which you can determine the approximate time allowed to heat up the system, assuming, of course, that you can remove the air from the system ahead of this time.

No. 890

WELDED PIPING FOR BUILDING HEATING SYSTEMS

By F. G. OUTCALT,[1] CHICAGO, ILL.
NON-MEMBER

NEW developments in the heating industry have led to increased efficiencies and economies in operation. In the modern systems of steam and hot water heating practically every item involved has been changed in some manner, with the notable exception of the piping. Until recently, lengths of pipe have been joined by cutting to length and threading. Now, welding is being used to a large extent for the fabrication of piping systems for heating, though pipe welding is by no means new, as it has been used in various industries for many years, particularly in the petroleum and chemical industries.

EFFICIENCY OF WELDED CONSTRUCTION

The term *efficiency* as used in this paper is intended to signify the degree of freedom from objectionable interferences with or the effects upon the purpose for which the piping system is installed, and involves the fittings, valves, or other appurtenances which might have an effect upon the flow of gas or fluid through the system.

Restrictions against the freedom of the flow of fluid or gas are eliminated where welded joints are used, and such welded joints are permanently tight, so that leakage losses are avoided. The danger of corrosion and leakage is greatly reduced, because corrosion does not take place at the welded joint. Furthermore, the welded joint is as strong as the pipe itself, and there is no necessity for increasing the pipe wall thickness to provide for the necessary strength at the roots of threading.

ECONOMY OF WELDED CONSTRUCTION

The economy of the welded piping system begins in the drafting room, due to the simplicity of welded construction. Standard pipe for the entire system may be purchased in single or double random lengths, depending upon the

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nature of the job. Material costs are lowered through the elimination of fittings since, with the exception of valves and their connecting flanges, these are made up on the job from standard pipe. This is particularly important in the case of headers, elbows, bends, anchorages and supports, where all fittings may be eliminated except at the valves. In some recent installations, valves have been furnished with short nipples welded to the valve body, so that no flanges were required, the valves being welded directly into the line. A saving in erection is next effected through the reduction of total weight of pipe and supports and the consequent cost of handling. Much scaffolding is eliminated since sections of the pipe may be fabricated on the floor and tied into the system at convenient points. The cost of insulating the pipe is lowered as there are no flanges or flanged fittings to be covered with molded insulation. Waste material is almost entirely eliminated because short

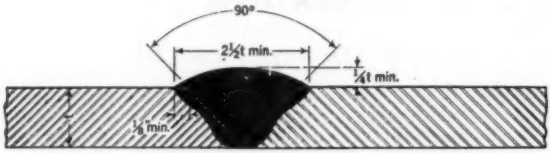

FIG. 1. OPEN SINGLE VEE BUTT WELD

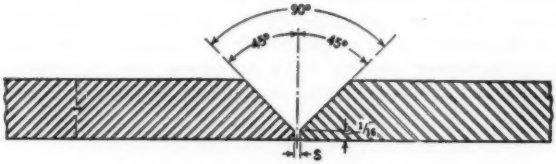

FIG. 2. PIPE BEVELED FOR WELDING

lengths of pipe can be used and no extra cast fittings are required. No delays are experienced because of alterations or additions which are not in the original plans, or in waiting for special fittings to arrive from the factory.

JOINT DESIGNS

The open single *vee butt* weld, shown in Fig. 1, is the standard welded line joint. It is easily and economically made and according to tests of many types of joints, it appears to be the strongest. Pipe to be joined by this weld is supplied by the manufacturer with ends machine-beveled, as shown in Fig. 2. The bevel does not extend to the inner wall of the pipe; a flat portion about 1/16 in. wide, for all sizes of pipe, is left to facilitate weld depth penetration and lining up. It also serves to strengthen the edge during transit and handling. When pipe is lined up for welding, a space *S* should be left between pipe ends to allow them to draw together as the cooling weld metal contracts, thus eliminating the possibility of overlapping edges or undue residual stresses. No allowance need be made for this spacing in design layout of a piping system as the overall lengths will be true within construction tolerances after welding. To add an extra factor of safety, the weld is usually built up, or

reinforced, about one-fourth of the wall thickness, as shown in Fig. 1. This reinforcement should slope gradually from the center down to the surface of the pipe alongside the weld. The width of the weld should ordinarily be about two and one-half times the thickness of the pipe wall. This type of weld is recommended for standard, extra heavy and double extra heavy piping, for all services, carrying pressures to which steel or wrought-iron pipe is subjected.

Tee fittings, crosses, and branch connections, either of equal or unequal diameter, can be fabricated from standard pipe by cutting with the blowpipe and welding to form the finished fitting. Templets are used for making the cuts accurately. Such fittings should be carefully prepared to permit proper fusion to the bottom of the vee and to present a smooth flow condition. One

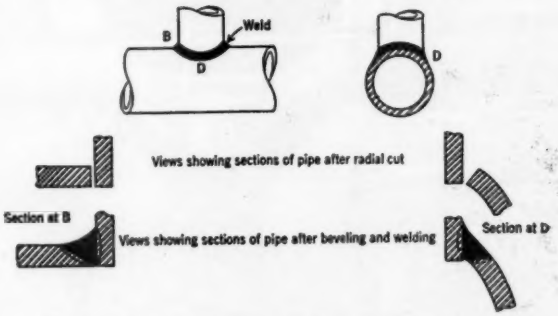

FIG. 3. DESIGN DETAILS FOR BRANCH CONNECTION

of the most desirable designs is obtained by inserting the branch into the hole down to the inner surface of the header wall. When the cut in the header has been properly beveled, this type of intersection presents a very good vee for welding. Fig. 3 illustrates this type of assembly. These two types of joint design are fundamental and are most frequently used in welded piping construction, although other designs are used occasionally for various fittings.

FITTINGS FOR WELDED PIPING

With recent rapid developments in the welding of pressure piping, there have been introduced a number of novel features which not only make for increased speed of erection, but also improve the strength, continuity and efficiency of the finished piping system. These include the forged high-hub welding flanges and forged reducing fittings. Forged high-hub welding flanges are simply forged steel flanges having a beveled hub of sufficient length so that they can be welded to the end of a pipe without any possibility of distorting the face of the flange. Forged reducing fittings are short lengths of pipe swaged down at one end, which can be welded into the line at any point where a change in diameter is required.

Another class of fittings designed especially for welded construction is the short-radius quarter and 180-deg. elbow. These short radius elbows are forged

from seamless steel pipe or tubing by special processes, and are of great service to welding erectors on account of the ease with which a section can be cut off and welded in to form an elbow of any desired angle. They have ample strength, are sightly in exposed piping, and greatly facilitate the application of insulating covering.

Fig. 4 shows the use of 12-in. extra heavy short radius bends in an installation designed for a working pressure of 200 lb per square inch.

WELDED HEADERS

Fig. 5 shows a typical welded pipe header fabricated by means of the types of joints previously shown. Its counterpart constructed with standard cast

FIG. 4. SHORT RADIUS ELBOWS IN 12 IN. EXTRA HEAVY LINE

iron flanged tees is also shown. The welding process is particularly adaptable to this type of piping construction. Headers are entirely fabricated from standard pipe, without the use of cast or forged fittings, unless flanges are used for connecting to branch piping or valves. Thus a compact design is obtained with great economy of weight and expense. Trouble from leaky gaskets is permanently eliminated. Delay in waiting for delivery of fittings, which sometimes cannot be ordered until boilers and other equipment are in place and lined up, is avoided. Due to the flexibility of welding processes, minor changes in design not included in the original plans can easily be made. Insulation can be applied neatly and at low cost to a welded header, due to the elimination of flanges. A welded header weighs much less than a similar one made up of cast fittings.

Another type of header, of more recent design, which has some very interesting features, is shown in Figs. 6 and 7. This is the main header for four 250-hp boilers for heating a large building containing approximately 50,000 sq ft of direct radiation, augmented by several unit heaters and a cen-

tral blast system. This header consists of two 15-ft lengths of 16-in. pipe welded together and blanked off at each end by pieces of ¾-in. flat plate welded in. The four 10-in. boiler leads are welded throughout from the long-hub, forged-steel welding flange and short radius bend at the boiler connection to the short radius bends welded to the header at the other end. All of the mains are taken off by means of short radius bends. These are welded directly into the header and perform double duty in providing outlets for the header and changes in direction through the use of a single fitting. The entire header, including all connections, was fabricated on the floor and hoisted into position in one piece. The space saving is notable, the over-all vertical dimensions being about 34 in., which compare with a minimum of 53½ in. required for flanged construction. The weight of the welded header is approximately 2,650 lb while the weight of a similar header made up of flanged cast-iron fittings would be about 10,400 lb or nearly four times as great. Another 6-in. outlet was later welded into this header.

ADAPTABILITY OF THE OXY-ACETYLENE PROCESS

The adaptability of welded construction to structural details of the building in which it is installed is illustrated by the following change of plans in the

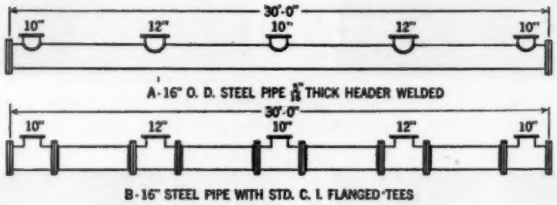

FIG. 5. HEADER DESIGNS

distribution floor that occurred during the construction of a large building, a view of which is shown in Fig. 8. As originally planned, the mains were to be exposed, and it is evident from the 16½ in. space above the tops of the windows that there is no room for an attic of the usual proportions for concealing the piping. Since this floor was to be used for offices, it was later decided that exposed piping was not desirable and that the pipe should be concealed if possible. The main and branches were successfully placed in a space of 14¾ in., after allowing for steel and lath and plaster for the hanging ceiling below the pipe. Since the largest main was 12 in. in diameter, leaving only about 1-in. clearance above and below the pipe, it was necessary that runouts to the branch risers be welded into the main eccentrically to provide for drainage of condensate. To insure accuracy, templets were used in making the cuts for these connections. This main was fabricated on the floor in lengths of 40 to 80 ft with short lengths of branches welded in.

The sections were raised and hung temporarily a few inches below their final positions by the use of long hanger rods. Space was thus provided for making the position welds and the main was later raised to its permanent location by screwing up the hanger nuts, after which the surplus rod was cut off. Fig. 8 shows the proximity of the piping to the ceiling. Fig. 9 shows

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FIG. 6. NOVEL HEADER DESIGN

FIG. 7. SHORT RADIUS BENDS WELDED DIRECT TO HEADER

FIG. 8. WELDED PIPING INSTALLED IN MINIMUM SPACE

FIG. 9. WELDED REDUCER AND BRANCH CONNECTION

FIG. 10. WELDED PIPING COVERED WITH INSULATION AND READY FOR THE HANGING CEILING

the details of a reducer and branch connection in this main on the floor before it was hoisted into position. The standard 45-deg bevel and the spacing of the pipe ends to permit thorough fusion and to provide for contraction of the cooling weld metal, are plainly shown. The appearance of this main after insulation was applied and ceiling steel was in place before lathing is shown in Fig. 10. A 6-in. main is shown in the foreground and the reduction from the 8-in. main, after a branch has been taken off, is shown in about the center of the illustration. The latter main was installed level on account of the small amount of space available, and all condensate was carried off by the eccentric branches.

EXPANSION AND CONTRACTION

Provision for expansion and contraction of welded piping is made in practically the same way as for other types of piping construction. Standard expansion loops are ordinarily used. Good practice in installing expansion loops is to spread them a sufficient amount by means of a jack, prior to connecting, so that they will be in a neutral position and not subject to any bending stress when the line is at operating temperature, if the line is to be

FIG. 11. WELDED EXPANSION LOOP

hot at all times. If the line is to be hot in winter and cold in summer, the loop should be spread half this amount so that it will be subject to but very slight bending stress either when hot or cold. Fig. 11 shows how an efficient expansion loop can be fabricated by welding through the use of short radius elbows.

PROCEDURE CONTROL

Successful results in the installation of a welded piping system can be insured only by the application of some definite regulations or methods that may be prescribed for the purpose of bringing together all of the essential elements for good workmanship. These embrace such elements as selecting the necessary materials of good welding quality, correctly designing the joints, properly preparing the parts for welding, predetermining the ability and knowledge of the welding operators, applying the most approved welding technique, and finally, properly testing the completed joints. This *procedure control* will not only insure safe welded construction, but also will make it possible to duplicate results at any place and at any time. Experience covering a period of many years has definitely proved the reliability of this method of application of welding, and has demonstrated that engineers and architects can use welded piping installations for all pressures and services with complete confidence of successful results if a well-established procedure control is followed.

DISCUSSION

W. C. BEVINGTON (WRITTEN): Today steam production is quite efficient and economical, but plants are continuously being penalized by inefficient distribution systems. Little thought seems to be given to the loss of steam through leaks in the piping system, as a majority of new building installations are still being fashioned on the screwed joint and companion flange principle. The piping division of the heating system has shown but little improvement since the time when the first threaded cast iron fittings were made. Many tons of pipe and fittings with screwed joints are actually built into the masonry construction of buildings with no provision for access to the piping.

I have observed leaks in flanged work where the losses were estimated at 200 hp and the plant was in daily use. The loss in radiation is also of some importance, due to the difficulties encountered in properly insulating these joints.

Too much dependence is put in the threaded joint to provide for expansion and contraction. Under high pressure and extreme temperature changes causing considerable expansion and contraction of flanges, bolts and threads, the average life of a gasket is extremely brief and under conditions of rapid fluctuating temperatures and pressures it is even more so. Due to the large amount of additional metal in the flanges, radiation losses are high whether the joint is covered or not. The most costly feature of such a joint is the tendency to leak.

The use of welded joint work requires proper expansion loops and bends to be designed as a part of the piping system and expansion is properly provided for without strains and movement in the threads.

The use of welding has brought out the long hubbed flange keeping the heat far enough away from the face of the flange to eliminate any warping and lends itself admirably to use with the proper expansion loops and bends.

With the introduction of seamless drawn fittings of neat appearance and practical shapes an installation can be made pleasing to the eye as well as efficient and economical. Heating system piping represents the highest type of welded piping con-

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struction and the application of procedure control assures high quality work. I believe there should be written into all specifications just how all welders should qualify and how the welding should be done.

Since the quality of the weld is no higher than the ability of the welder, an organized effort and campaign should be carried on to educate mechanics into the field. Rigid standards should be set for testing work and all should be enforced.

One of the distinct advantages found in a welded installation is the minimum scaffolding required on the job and the comparative ease of handling pipe, due to the elimination of heavy, cumbersome fittings; together with the elimination of heavy pipe tongs and wrenches.

The use of heavy tongs while working on scaffolds presents a hazard to the workmen which is eliminated in welding work, as the torch easily can be used with one hand. The time consumed in making a welded joint is less than for a screwed joint, and the saving of time is reflected throughout the entire installation.

Welding is decidedly advantageous in close quarters where space is limited and solves the problem of concealed piping. In headers where it is virtually impossible to get the number of leads desired in a close space, using fittings, etc., a welded header can be made up with positive assurance that everything is on center.

Welding has the advantage of unrestricted passage from mains to branches. Each fitting in a screwed joint job causes an eddy current to be set up which acts as an obstruction to the free flow because the inside edges of the pipe are not continuous as they are in welding. Operating experience in pipe line work has shown that a welded pipe has somewhat greater carrying capacity than other types of the same diameter and the reason may be traced directly to reduced friction of flow.

Summing up, the welded joint is efficient and practical. In a welded boiler which had been damaged by an explosion from internal causes it was found that the fractures were in the body of the steel plate, the nearest fracture being over 4 in. from the welded seams. This is conclusive evidence of the strength of a welded joint.

Our records show that the cost of the entire installation using the welded joint is approximately 10 per cent lower than that with the screwed joint. The saving in the cost of covering a welded piping system is quite an item due to the elimination of the special shapes for fittings and the labor involved.

Welding is here to stay and must be provided for. It must be set upon a high plane and constant vigilance exerted to maintain the required standards. I would suggest the writing into all specifications, provision for a rigid procedure control and rigid qualification tests for the welders.

JOHN HOWATT (WRITTEN): The welded type of joint which is rapidly supplanting the screwed joint in steam piping systems causes no difference in design, in fact the work of the draftsman is simplified because he can, to a large extent, disregard linear measurements.

The paper makes no mention of comparative costs as between the screwed and welded joint job for steam lines. In arriving at comparative costs consideration must be given to the kind of pipe and fittings to be used in the screwed pipe joint. For example: a screwed joint job may be of standard weight steel pipe with standard fittings, standard weight steel pipe with extra heavy fittings, extra heavy steel pipe with extra heavy fittings, or wrought iron or copper bearing steel pipe with the above fittings. Whether the installation cost of the welded joint job will be lower than the installation cost of the screwed joint job will depend to a large extent on the position or location in which the piping will be fabricated and the cost of the fittings used. On an ordinary large steam heating job we have found in our Chicago school work that when using standard steel piping with standard weight screwed fittings the installation cost will be lower for the screwed job on all piping below 4 in. in size but will be higher than the welded joint job on all piping over 4 in. in size. This, of course, will vary with conditions.

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Where experienced competent welders are difficult to obtain the contractor must estimate a higher figure for the work than he would for a standard screwed pipe job. Where experienced competent welders are obtainable, however, contractors have costs well determined so proper competitive bids can be obtained.

Information on the efficiency of the welded joint on copper bearing pipe would be of interest.

The paper points out that a factor of safety is provided by building up a welt of iron on the outside of the joint ¼ the thickness of the pipe wall. I do not believe the ridge is important as it is my opinion that the weakest part of the joint is usually at the inside. Much depends upon the care and skill used by the welder.

The long life of the welded job and its freedom from leaky joints and repairs are important advantages. A steam piping job in a tunnel, which several years ago was giving constant trouble from leaky joints, was converted into a welded joint job and there have been no leaks or repairs since. Experiences like this favor the growth of the welded joint steam heating pipe jobs. However, there are locations where the screwed joint is easier to install than the welded joint and in smaller sizes it is still a lower cost job.

G. D. LUCE (WRITTEN): I would like to see this subject carried out to its logical development; which is, the welding of all the piping in a steam heating system with the possible exception of the individual radiator connections.

A pipe installation is needed which will last a reasonable number of years without excessive repair and replacement costs.

The majority of failures occur where the pipe wall has been weakened by the thread, and it is logical to assume that the elimination of threads would obviate much of this trouble.

Little if any progress is being made in welding complete jobs. Some development should be started at once on the part of the manufacturers of welding apparatus and the heating and piping contractors, to the end that heating contractors would equip and tool up their shops, and through their own experiments and efforts acquire sufficient experience so that specifications could be issued calling for a completely welded job with reasonable expectation of getting some intelligent bids from competent contractors.

Eventually we will be able to install a completely welded job of piping for the same cost or less than a corresponding job of screwed or flanged piping.

C. W. OBERT (WRITTEN): I endorse all that the author says on the advantages of the welding process.

Probably the most potent influence in this welding development is the economic phase. It was due to savings in time and cost that welding was first tried out in pipe erection and repair work. With a gas cutting outfit, which is so readily adaptable to welding, accessible on the job, welding came to be used in emergency or for unusual details of construction.

As experience was gained with this process, the urge for its use for piping erection changed from the economic phase for the reason that welded piping was found to be *better* for many reasons mentioned by the author. Operators of both high- and low-pressure plants now have sufficient evidence to influence the careful thought of both designing engineers and piping contractors.

Experience has shown conclusively that all that is necessary for the success of welding when applied to piping erection is proper procedure control to insure conditions that are conducive to good welding, and obtaining the necessary skill in the welding operators.

Fusion welding also is invading other fields of construction with equal success, such as, in structural steel, large pressure vessels, machine frames to replace castings

and for all classes of repair work. While it is true that there is often a marked saving, generally there is some outstanding advantage in the use of welding that governs, for instance, the elimination of noise in fabricating structural steel, or the insurance of absolute tightness in high-pressure petroleum vessels. The welding process has proven its merit, and both the engineers and the contractors should prepare for its predominant use in this field.

W. H. MCCAULLY (WRITTEN): On two jobs we were able to make a substantial saving by having the larger size pipes (over 2 in.) welded. This to the owner is the most definite proof of economy.

Not only can minor changes be made easily, as the author has pointed out, but also major changes in piping systems are possible. Often changes occur in the operation of a business while a building is being constructed and it is, therefore, necessary to make important changes in the building which were not contemplated in the original design.

The main feeder shown in Fig. 6 running out in a diagonal direction, was not decided upon until the other work was practically in place, but it was added as easily as though it had been considered in the original design. Connections of this type that affect the heating system can be made at negligible cost when the system is welded.

We had occasion to tap a 16 in. line with a 1 in. line. This was done easily by welding, whereas it would have been most difficult in a job of flanged fittings.

H. M. NOBIS: In my opinion welding pipe less than 1¼ in., especially risers, is not good practice because if you cut these sections (split the pipe), you will see some volcanic eruptions protruding toward the inside. I cannot see how even procedure control can control this.

Welding may be more practicable if other alloy steel pipes which are lighter are used because a man can weld faster than he can with the heavier pipe.

J. C. FITTS: *The Heating and Piping Contractors National Association* has been working very diligently on this subject for the last two years and results of its studies will be published soon.

The Association's Standard Specification for Welding will save engineers a great deal of labor, and by standard courses, steam fitters are being trained in a uniform manner so that welding can be specified with perfect surety of quality work.

Having recognized standard specification and being able to deal with responsible contractors employing properly trained men, the close inspection becomes unnecessary because if there is any error the test will pick it up.

JULIUS ROEMER (Cleveland): The responsible contractor obtains his men from the various steam fitters locals.

An effort is being made in Cleveland to ask labor to supply certified welders, who must be able to pass this welding code test which will probably come out in the specification by the various engineers. Labor is to appoint one representative, the heating and piping contractors another and a neutral person to form a test committee for testing and certifying welders. In case a local organization can't supply men they will be obtained from other locals. If they can't supply trained union steam fitters they will give us capable commercial shop men. Therefore we can give the engineer and owner a real, certified job.

L. A. GREEN (Pittsburgh): The Steam Fitters Local Union of Pittsburgh has made every effort possible to have their membership qualified to meet the demands of welding in the industry. Last year we registered 90 men at Carnegie Institute of Technology for instruction under Prof. Dibble.

The Heating and Piping Contractors Association requires that each member take instruction in welding and about one-third of them have become welders.

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The paper's criticism of the journeymen is too severe in my opinion.

It is the purpose of the steam fitters local unions throughout the country and the United Association to have their journeymen steam fitters qualify for welding.

MR. SULLIVAN (Chicago): We have a school and a laboratory in Chicago which is kept up by the *National Association of Heating and Piping Contractors* in conjunction with the steam fitters organization. They work in close cooperation and through the employers they are giving us all the assistance possible.

MR. BRADLEY (Mellon Institute): Regarding the specification for welding, I wonder if any effort has been made to specify the welding wire which is to be used for the purpose.

MR. FITTS: In the specifications of the *Heating and Piping Contractors National Association* we have not specified any wire by trade name but by content.

CHAIRMAN ROWLEY: Mr. Outcalt will close the discussion.

MR. OUTCALT: Chicago is one of the pioneer cities in combining the work of the local steam fitters union with the Board of Education, which has supplied quarters, apparatus and gases at a nominal fee to union members taking this course. As a result there are many qualified welders in Chicago.

Certification of welders may lead to rather questionable results; that is the certification by any municipal or state body. Therefore, we strongly emphasize the desirability of certification of welders by qualification tests preceding the job or by qualification records kept by the contractor and available to the specifying architect or engineer.

As for economy of welding, don't get the idea that you can save 10 per cent on every job. You can not. Some jobs may cost more but you will probably have better jobs when you finish.

An article by Konrad Meier in *Heating, Piping and Air Conditioning,* revealed that on joints which were well made the frictional coefficient was zero and also on joints of sizes larger than 2½ in. in diameter, the frictional coefficient on the worst joints pictured was no greater than the frictional coefficient of a screwed elbow.

Little is known about the possibilities of welding complete jobs, although there have been complete jobs welded. In a heating installation in the Northern States Power Building in St. Paul, a local contractor welded the complete job right down to the radiator valves. However, I wouldn't recommend that in all cases nor for all contractors, at this time. In fact, I would caution against it until the contractor, as well as the engineer, is willing to rationally study welding and its possibilities and forget many of the out of date practices that have grown up because of the limitations of the older methods of construction.

Although design does not have to be changed where welding is substituted, there are instances involving the use of screwed connections to take care of expansion, where design changes can be effective in reducing the operating as well as the installation costs of the systems.

As an example, in coming from a main and going to a riser, you probably want slope in the pipe. If using ordinary fittings you probably will have to go around 2 bends at least in order to have the pipe slope from the main to give proper drainage. It is comparatively simple when welding to come directly from the main at any desired angle by cutting the elbow to the proper angle, running directly to the riser and providing sufficient length to take care of expansion by changing the location of the main connection if necessary rather than by putting in a lot of extra pipe and fittings.

Another welding economy lies in the possibility for using lighter wall pipe.

Most all pipe that is made today can be welded, including brass, copper and cast iron, as well as wrought iron and steel of all grades.

No. 891

AIR POLLUTION FROM THE ENGINEER'S STANDPOINT

By H. B. MELLER,[1] PITTSBURGH, PA.
NON-MEMBER

SMOKE abatement is the term generally applied to efforts to reduce the volume of black smoke emitted from chimneys as a result of incomplete or improper combustion of fuel. Entirely aside from the damage to health and property resulting from the pollution of city air by the solids and gases issuing from chimneys, inefficiency in the burning of fuel is a direct money loss. It would be no exaggeration to say that 20 per cent of the present fuel bill of this country is wasted by preventable inefficiency.

The smoke nuisance is as old as the use of coal, and agitation against it has been in proportion to the discomfort caused. At present 125 cities in the United States, over 30,000 in population, have anti-smoke ordinances. Special surveys of conditions have been made in New York, Chicago, St. Louis, Salt Lake City, Boston, Baltimore, Pittsburgh and other cities.

ORIGIN AND COMPOSITION OF SMOKE

Fuels that contain little volatile matter, such as anthracite and coke, will burn without appreciable visible smoke, although ash is carried off in proportion to the draft. On the other hand, semi-bituminous coals, to a degree, and bituminous coals high in volatile matter, may be bad smoke producers, as may liquid fuels under some conditions. As a rule, the amount of volatile matter may be considered an indication of the smoke-producing nature of coal. But volatile matter itself varies in composition from that containing gases and compounds that are not difficult of combustion in an ordinary furnace without the production of visible smoke, to that which can be burned without producing dense smoke only in carefully constructed and operated furnaces.

Combustion and Its Products

Theoretically, the process of combustion is simple, being a chemical union of the combustible material of a fuel with oxygen from air, the combination

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resulting in the development of heat. In practice, however, many difficulties are encountered which tend to complicate the process, and it is in overcoming them that care is required in the construction and operation of furnaces. The number and nature of the difficulties vary considerably with the composition and fineness of the fuel, the type of furnace, the method of firing, the supply of draft, the amount and composition of impurities in the fuel, and the variable demands for heat or power. Briefly stated, the stages of combustion are:

1. Absorption of heat.
2. Distillation and combustion of the volatile matter.
3. Combustion of the remaining or carbonaceous part of the fuel.

The difficult part of the process is to secure complete combustion of the products evolved in the second stage, before they strike any cooling surfaces that will reduce their temperature below the ignition point. The requirements of complete combustion are:

1. Air in sufficient quantities must be admitted at the proper time and place.
2. Thorough mixture of the air with the gases evolved, which involves an element of time, requiring sufficient space in the combustion chamber.
3. The temperature of the gases must be maintained above their kindling point until combustion is complete.

If these conditions are met, the products of combustion will be carbon dioxide and water vapor, plus sulphur dioxide (SO_2), sulphur trioxide (SO_3), a small amount of hydrogen sulphite (H_2S), various sulphur-containing organic compounds, and in addition, such solid particles of ash and carbon as may be taken out by the draft. Only a haze will be visible coming from the stack. On the other hand, incomplete combustion will result in the production of visible smoke in smaller or larger quantities.

In installations where the temperature and draft are comparatively low, and the methods of firing irregular—as in the domestic furnace—the soot contains a comparatively large percentage of combustible matter (carbon, hydrogen and tar) with a small percentage of ash. Where the furnace temperature and draft are comparatively high, small amounts of coal being fired at relatively short intervals, the percentage of combustible is much lower, but larger amounts of ash are carried off.

In industrial cities, the sootfall amounts to hundreds of tons per square mile per year. This deposit may contain as much as several per cent of tarry matter and 20 to 30 per cent of fixed carbon.

EFFECTS OF SMOKE [2]

In a consideration of effects, there must be taken into account all of the products of combustion—not merely visible smoke.

Meteorological Aspects of the Problem

While the elementary gases—nitrogen, hydrogen, oxygen, and the gases of the argon group—are practically fixed in amount with respect to one another at

[2] The information presented in this section and also in the next two sections has been gleaned from the results of studies made under the auspices of Mellon Institute during the period 1911-14 and in 1923 and 1930.

any given level, the compound gases, such as water vapor, ammonia, ozone, and carbon dioxide gas are varying constantly, not only with respect to the atmospheric conditions, such as temperature, pressure, and humidity, but also with respect to place. This is especially true of gases generated in the process of combustion, which may be almost unknown in thinly inhabited districts, but which must be taken into account in the atmosphere of large cities.

Suspended Matter. The gases of the atmosphere hold in suspension considerable quantities of solid and liquid particles. The latter consist principally of condensed water vapor, forming fogs and clouds, which may hold in solution substances of various kinds, including some acids. The solid particles are meteoric dust, finely divided mineral matter or soil from the surface of the earth, pollen from plants, microscopic organisms such as bacteria and moulds, and the residues of combustion.

Dust Layers in the Atmosphere. There exist in the atmosphere three well-defined dust layers, of which the lowest, less than a mile in thickness, is of primary importance, because in it are carried on nearly all human activities. This dust layer consists of comparatively coarse particles caught up by the turbulent surface wind. Its density depends upon the relation between the rate of supply and the rapidity with which the particles can be carried away or distributed throughout the atmosphere. The least dust is found in sparsely inhabited regions covered with dense vegetation. In large cities, especially where bituminous coal is burned, the discharge from numerous chimneys produces the densest dust or smoke layer. Diurnal convection carries fine dust to a height of nearly three miles, and the convective action of storms carries it to the base of the stratosphere—the portion of the atmosphere lying above the region of convection—at a height of about seven miles.

The effects of atmospheric impurities upon meteorological conditions depend not only upon the size of cities and the rate of discharge of their impurities into their atmospheres, but also upon the rate at which the vitiated atmosphere is carried away. They vary, therefore, with geographical position, season, average wind velocities, and topography.

Smoke aids in the formation and prolongation of fogs. Condensation of atmospheric moisture takes place only when the saturation temperature has been reached, and then only upon free surfaces, such as the surfaces of suspended dust particles. In the city, fogs become mixed with smoke to such an extent that their color is frequently changed from white to brown, or even black, and their density is so increased that they are almost opaque to sunlight. And, of course, the lighter the wind the greater the density. Then, too, the absorption by soot particles of moisture from the fog so increases the weight of such particles that they quickly settle to the lower atmospheric levels. Oily or tarry substances also form a coating on the fog particles and retard evaporation. Also, in the presence of sulphuric acid, evaporation of the fog particles is retarded. Furthermore, the increased opacity of city fogs prevents penetration of heat rays from the sun to any considerable depth, so that fogs persist materially longer in a large smoky city than in the suburbs, and still longer than in the country.

It is obvious that the observed hours of sunshine are less in large cities than in the suburbs or in the country. However, the sunlight which does

filter through is deficient in rays of the shorter wave-lengths, thus reducing materially, and at times cutting off entirely, the beneficial ultraviolet radiation.

Effects of Smoke on Stone

The products of combustion of fuels that are injurious to stone may be divided, roughly, into two classes: (1) those that soil and render the buildings unsightly, that is, carbon, tar, ash and their various combinations; and (2) those that corrode and in general aid in the destruction and weathering, that is, sulphuric acid (H_2SO_4), and sulphurous acid (H_2SO_3), hydrogen sulphide (H_2S), hydrochloric acid (HCl), ammonia, the organic acids, etc.

The action of carbon and ash, when not associated with tar, is not extremely objectionable, as the simple process of brushing will remove either. When, however, they are associated with tar, especially in the case of soot, they produce the most objectionable kind of dirt. The tar causes the soot to adhere firmly to any surface with which it comes in contact, literally covering the object with a coating of black paint and penetrating into the porous structure. This coat is not readily removed because it has adhesive properties and also because it is insoluble in water. Drastic measures, such as the use of solvents, scouring, or both, frequently are necessary. Steam cleaning has been used recently for this purpose.

Of the corroding agents, sulphuric acid is the most important. Sulphurous acid, while of itself not particularly active, is changed rapidly by oxidatiton in the air to the far more corrosive and injurious sulphuric acid. Hydrochloric acid is of less importance; ammonia and hydrogen sulphide are practically without corrosive action on stone, but contribute to the corrosion of bronze and other exposed metals.

It might not seem that the amount of acid coming from the chimneys of a large city would be sufficient to do much harm. However, coal as fired contains an average of at least one per cent sulphur, or 20 lb. per ton of coal, and, say, 75 per cent of this constituent, or 15 lb. per ton of coal burned, escapes into the atmosphere. If, for example, this is considered as sulphuric acid, it would be equivalent to 46 lb., which, if it all were allowed to act on limestone ($CaCO_3$), would destroy 47 lb. and produce about 80 lb. of gypsum ($CaSO_4.2H_2O$).

Part of the acid as it leaves the chimney is occluded in the soot and part goes out in the form of gas with the other products of combustion. That which escapes with the flue gas is dissolved by rain and will corrode any material with which it comes in contact and upon which it can act, but it is soon washed off. That occluded in the soot is more injurious, as the tar causes firm adherence to anything with which the soot may come in contact. This acid is not readily washed away by rain and in all probability, if it is adhering to corrodible material, the action is complete.

Stones that do not contain the carbonates of calcium and magnesium—granite, gneiss, and sandstones in which the grains of sand are cemented by substances other than the carbonates—are but little affected by the corroding agents derived from combustion. They are, however, soiled by soot, and the necessary cleaning by wire brush and detergent chemicals, a sandblast, or other drastic method, is not only costly, but injures the stone itself. All rock contains water, the greater the percentage of porosity, the more water. After removal from

the quarry, this contained water, as well as any which is absorbed by the rock after quarrying and which becomes saturated with the various substances of which the rock is composed, gradually comes to the surface and is evaporated, depositing the dissolved minerals in the form of a hard, compact crust. The removal of this protective crust hastens the weathering of the stone. Furthermore, the stone is soon as dirty as before.

The action of sulphuric acid on stones containing larger or smaller amounts of the carbonates of calcium and magnesium—limestone, dolomite, dolomitic limestone, and sandstones with a calcareous cementing material—is, for the most part, the action of this acid on these carbonates. This chemical action involves a change in volume. Still more important than the volume change is the pressure developed by the sulphate crystals growing in definite oriented directions. This is apt to occur in all kinds of stones; even granites are not exempt. The stone becomes more porous, loses its cohesion, and disintegrates with much greater ease under the action of the weather.

Mortar, because of its rather porous structure, and because it is in process of chemical change for some time after it is placed in the wall, is particularly subject to the action of the atmospheric acids. This will be evident upon examination of the mortar in old buildings.

What has been said for stone applies equally to concrete. The griming of the surface is just as serious, and the action of the sulphuric acid is at least as detrimental, if not worse, for it reacts upon the aluminate of the cement, forming a complex substance the volume of which is three times that of the original aluminate. Moreover, both this complicated aluminate and the gypsum formed by interaction with the lime in the cement grow under favorable conditions into crystals of appreciable size, which, acting like millions of tiny jacks, produce a tremendous pressure in the aggregate.

Effects of Smoke on Metals

Soot as a Coating. Soot that does not contain injurious constituents may, in fact, act as a protective coating. However, this coating is not uniform; and where metallic surfaces are intended to be of decorative value, the effect is spoiled, if not entirely destroyed, by the mottled soiling. The remedy in such case is simple—wipe off the dust.

When the surface is covered by soot containing tar, or soot and tar separately, oftentimes scouring is necessary to clean the surface, but this results in injury to the metal through abrasion. If the soot is allowed to remain, any intended decorative effect is destroyed in this case also.

Effect of Hydrogen Sulphide and Ammonia. These two gases are to be considered as definite sources of corrosion of metal work and as the cause of the staining of certain types of paints and of bright metal surfaces. Of the two hydrogen sulphide is the more active and troublesome, since the metallic sulphides which are formed first are easily oxidized to sulphates in moist air, and these salts accelerate further corrosion. It should be emphasized that under ordinary conditions the quantities of hydrogen sulphide and ammonia in the atmosphere are so slight that they do not appear to be major causes of corrosion. Because of their possible catalytic action, however, they are not to be disregarded.

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Sulphur Acid. In estimating the possible damage to a municipality from the effects of the sulphur acids, it must be borne in mind that not only sulphur from fuel burned in that municipality must be considered, but also that which originates from other communities in the same district, much of which is brought in by air currents, just as a considerable amount of that from the city is carried out, to do its damage elsewhere.

Finely divided carbon has the power of occlusion to a high degree, a fact which makes possible the retention of some sulphur acids. Since corrosion occurs only in the presence of moisture, soot also aids corrosion by retaining moisture. Iron oxide has this power of occlusion, although to a less degree than does carbon; and when this oxide is formed, destruction of the iron is hastened by absorption of acid from the air.

When soot containing tar comes in contact with a metallic surface, the tar causes it to adhere more or less firmly. Thus the occluded acids, principally sulphuric and sulphurous, are brought into intimate contact with the metal, affording a much better opportunity for corrosion to take place and more quickly than if the same amount of acid dissolved in rain water were to come in contact with the metal.

The rapidity and completeness of corrosion by this combination are aided further by the fact that any free carbon that the soot may contain is electro-negative, while the metals are electro-positive. Thus galvanic action takes place readily in the presence of the acids and water.

Corrosion of the Metals. The destruction of iron is most noticeable, for it is the metal in most common use, and it is more readily corroded than the majority of metals. Iron remains unchanged in dry air and in dry, inert gases. However, when the air is moist, and more especially when it contains acid vapors, the iron rusts (oxidizes), forming hydrated oxide of iron. It is attacked by dilute sulphuric and hydrochloric acids with the liberation of hydrogen. Action of any of the corrosive agents is hastened greatly by any arrangement of substances which makes electrolytic action possible. The corrosion of either protected or unprotected iron work is much more rapid in city air containing the sulphur acids and other corroding agents than in the country. Investigations of iron alloys show that small quantities of some elements, when alloyed with the iron, materially reduce the rapidity of corrosion by the various corroding agents.

Zinc is acted upon readily by most dilute acids, the mechanism of corrosion being similar to that of iron. In moist air, a thin protective coating of a white, basic carbonate is formed; this coating, being soluble in sulphuric acid, is destroyed readily when the air carries acid vapors. Iron protected with a coating of zinc corrodes more rapidly, once the coating is broken, than either metal alone, because of electrolytic action.

Tin is little affected by the acid-containing air of smoky cities. However, where it is used as a protective coating for iron, corrosion is hastened by galvanic action, especially when part of the coating is worn away, exposing the iron as well as the tin. Where tin plates are soldered together and the soldering is not perfectly done, corrosion is aided greatly by electrolytic action.

Copper is acted upon very slightly by dilute hydrochloric and sulphuric acids. In moist air accompanied by acids, it becomes coated with verdegris. Both

copper and brass are particularly sensitive to the action of hydrogen sulphide. Copper and many of its alloys are attacked readily by both ammonia and nitric acid, which are present always in the air, although in small quantities.

Lead is not corroded appreciably by air, although its surface dulls on exposure, owing to the formation of oxide. Dilute hydrochloric and sulphuric acids have little effect on the metal, and that effect is beneficial in a way, especially in the case of sulphuric acid, as lead sulphate is quite insoluble.

Aluminum is unaffected by corrosion in a smoky atmosphere. Exposure of this metal under such conditions for as long as twenty years showed no appreciable deterioration. The surface accumulation of dirt may readily be washed off.

Effects of Smoke on Outside Painting

Particles of carbon deposited upon paint apparently tend to increase the life of the coating, since the ultraviolet light which affects the films is absorbed at the surface.

Sulphur dioxide is particularly harmful to a paint film when high percentages of carbon dioxide are present and when high humidity conditions exist. Paint films applied under such conditions nearly always dry with tacky surfaces that are subject to contamination by atmospheric carbon and dust, reducing the protective efficiency.

Paint darkened when applied to test fences, erected in different localities in the United States; the tarry matter from deposition of soot seemed to be thoroughly incorporated in the film. Paint applied over a soot-coated surface was attacked by the soot beneath as well as by that deposited on the exposed surface. Frequent washing of paint coatings is necessary, and paint must be renewed in much less time in a smoky atmosphere than in one comparatively free from smoke.

Effects of Smoke on the Interior of Buildings

Fine soot, as an impalpable powder, will go anywhere and everywhere that air does, and the tarry matter with it causes much of it to adhere firmly to any surface with which it comes in contact. Changes in atmospheric pressure produce changes in the air currents in shut-in or confined places, carrying the soot to the most unexpected corners. A common and striking example of this is the framed picture, with little dark sprays of soot along the white mats or on the picture itself, where the air currents have reached the space behind the glass.

The most noticeable damage of the soot is in impairing appearances and causing an enormous amount of extra, expensive and unnecessary work and care in keeping working and living quarters in such condition as is required by people accustomed to the common standard of living. While in the interior of buildings, damage by the sulphur acids is not so marked as in the case of material exposed to the weather, they undoubtedly cause considerable damage to delicate fabrics.

For the architect or decorator the market does not offer a paint or wall covering that is not susceptible to damage and the life of which is not shortened

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by soot. And the possibilities of the choice of colors for interior painting are limited appreciably. Wall paper must be cleaned more often in a smoky city; its life is much shorter than in a clean atmosphere. Lace curtains and hangings are soiled easily by soot and must be washed frequently. Soot is readily ground in on a floor rug. Altogether, the cost of upkeep of paints, decorations and furnishings is from two to five times as much as in a clean city. The cost of illumination is increased in a smoky city, not only by absorption of the light in its passage through the atmosphere, but also because of soot coating skylights, windows, lighting fixtures, etc.

Air Conditioning. For a building in which it is desired to have cleaner air than that of the outside—as for dry cleaning, laundering, food manufacture, etc., or in a department store, office building, hospital, school, club or elsewhere —air conditioning is applicable. Incoming air may be filtered or washed or both to eliminate such solid particles as can be removed by the process and to supply cleaned air of a predetermined humidity.

Effects of Smoke on Vegetation

The general opinion that smoke injures nearby vegetation is justified, as evidenced not only by the external appearance of many of the plants, but also by their internal appearance as shown by the size of annual rings and by lesions in leaves.

Experiments have shown that, especially in the case of pines, when soot is applied in small quantities over a considerable period of time, measurable injury follows. There is some clogging of stomata by carbon in the soot; but the injury done by the soot of smoke is due, probably, chiefly to the accompanying ash, tar and gases. Sulphur dioxide, when present, is absorbed by the leaves and forms sulphuric or sulphurous acid, which are toxic in the tissues of the leaves. Ordinary field and garden crops do not thrive in a very acid soil. Smoke is, therefore, injurious to vegetation.

PSYCHOLOGICAL ASPECTS OF THE SMOKE PROBLEM

The smoke palls of industrial centers have both direct and indirect effects upon mental well-being.

Direct Influences. The constant inhalation of poison-laden air probably results in a gradual process of absorption by the human system of the poisonous products of combustion. This insensible intake may not give rise to any definitely recognizable acute disorder or specific disability. But the process of slow poisoning may eat away, insidiously, like a mild canker, at vital tissues, making it impossible for body and brain to function at their points of maximal efficiency.

Indirect Effects. The indirect effects of smoke upon the human organism are traceable to the various meteorological states due to the products of combustion which get into the atmosphere and which, in turn, directly influence the body and mind. These are, particularly, sunshine, clouds, humidity, fogs, temperature, electrical potential and luminosity. It has already been shown that smoke aids in the formation and prolongation of fogs, that it reduces the amount of sunshine, particularly in the shorter wave-lengths considered so beneficial, decreases the limit of visibility and affects the diurnal temperature.

The action of these smoke-induced conditions upon the human being may be considered largely in terms of sunshine dynamics. On the one hand are the influences of positive states of sunshine: brightness, luminosity, warmth, dryness, the presence of the infrared, visible and ultraviolet rays, and high electrical potential. On the other hand are the effects of the negative states: gloom, darkness, cloudiness, fogginess, rain, moisture, cold, and low electrical potential.

On clear, dry, sunny days, persons are more active, aggressive and enterprising. On the other hand, dark, humid days are devitalizing, and therefore lead to inaction. Depleted vitality and nervous exhaustion are inhibitory of action.

HYGIENIC ASPECTS

In hygiene, the influences of smoke are both direct and indirect—direct from inhalation and indirect because of meteorological states resulting from atmospheric impurities. The psychological and hygienic effects of smoke are so closely related that it is impossible to draw clearly any line of demarcation. Bodily states normally influence mental states, and mental states, in turn, normally influence bodily processes, particularly the functions of the glandular, circulatory, sexual and neural systems.

Inhalation of Dust. When it is remembered that the average adult breathes about 30 cu. in. of air in each inhalation—equivalent to approximately 37½ lb of air per day, or about seven times the weight of food and water consumed—it does not seem an exaggeration to say that more persons are devitalized, disabled and poisoned by the impurities contained in smoke-polluted air than by the noxious ingredients in food and water. Not only do these air-borne impurities irritate the sensitive membranes of the eyes, nose, throat and lungs, and thus aggravate or cause diseases of these organs, or produce collapse of their sensitive tissues, or increase their susceptibility to acute respiratory diseases; but poisonous compounds also enter the gastro-intestinal tract and may cause nausea, vomiting, diarrhea and systemic poisoning.

A part of the human waste caused by smoke-polluted air certainly is due to the irritation caused by the solids. The functional efficiency of the visual, naso-pharyngeal, pulmonary, gastro-intestinal and neural mechanisms may be more or less disabled by such solid particles. But it is likewise presumptively possible that irritating, acrid soot particles and poisonous smoke compounds may, by slow, insidious action, become factors in depleting potential reserve, leaving open the way for sickness, or even premature decay and untimely death.

Relation of Smoke to Diseases of the Respiratory Tract. It is not surprising to find that the burden of proof has been sought in the relation of smoke to diseases of the respiratory organs, because the lung tissue is necessarily first invaded by the smoke content of the air inhaled. Accordingly, the majority of observers have based their studies upon the incidence of lung disease in smoky industrial cities.

Pulmonary anthracosis (not in coal miners) is distinctly an urban disease and is proportionate to the smoke content of the air. Although small quantities of carbon deposit in the lung may remain without harm, yet the quantity accumulating in the dweller of one of the larger cities has an accompanying

greater or less fibrosis, impairing the elasticity as well as altering the functional capacity of the organ.

In tuberculosis and granulomatous conditions in which the reactions are centered chiefly in focal lesions of the tissues, the anthracotic condition is either entirely passive, or is active in assisting healing, in that it is an additional stimulus to fibrosis and encapsulation, and in that it aids in the localization of the process through the obliteration of the lymph spaces. In acute inflammatory conditions where the lymphatics are important for proper resolution, anthracosis becomes seriously detrimental, because of the obliteration of these spaces.

Pneumonia, on the other hand, is a catarrhal condition, and a predisposition for it may be prepared by the irritation of the mucous membranes with foreign substances. The survey of the Mellon Institute (1912-14) showed a strikingly close relationship between the pneumonia death rate and the smoke content of the air as charted by wards.

Effects of Smoke-Induced Weather States. The amount of blood and the percentage of hemoglobin are increased by sunlight and decreased by darkness.

Smoke intercepts the sunrays of the shorter wave-lengths. These shorter waves, in addition to their effect in restricting bacterial growth, stimulate the powers of resistance to bacterial infection by their action on certain fatty substances in the skin.

Rickets, and other disturbances of the metabolism of lime salts very important to health, result when the short or ultraviolet rays of the sun are excluded from the body. When the nutrition of the body is thus vitiated, it becomes more susceptible to bacterial infection.

Soot has a definite bactericidal action, probably due to the action of its contained germicidal acid and phenols; it does not form a favorable nidus for the collection and distribution of bacteria. On the other hand, soot, as it occurs in smoke, clouds, fogs and as a non-transparent covering for streets and buildings, protects micro-organisms from the destructive action of sunlight.

THE RESPONSIBILITY OF THE ENGINEER

It may be asked why so much space has been taken to describe the effects of air pollution. The answer is that the responsibility for the development and application of corrective measures is with the engineer, and he will be better able to judge what is necessary if he has a knowledge of the havoc wrought by the impurities introduced into the atmosphere from equipment which he has designed, installed or operated.

To say that power or heating equipment satisfies a local anti-smoke ordinance is begging the question, because he knows that, in the nature of things, the law must follow development. A smoke abatement enforcement officer may suggest improvements, he may take advantage of possibilities that exist, but he cannot make any regulations impossible to be met by equipment already on the market.

As scientific research demonstrates how to use fuel more economically, and as *smokeless fuel* (especially coke and gas) and electric power come into more

general employment, advancement in effective smoke abatement will become more rapid. However, it has seemed to the writer that the heating and ventilating engineer has not given adequate encouragement to the development of combustion engineering as a specialty. As is generally known, there is a dearth of combustion engineers, due to the lack of instructional facilities for their training in this country.

A great deal has been done in the matter of smoke abatement through the co-operation that exists between the engineers and local bureaus of smoke regulation. Manufacturers have listened to suggestions made. There are stokers, powdered coal burners, *smokeless* boilers, secondary air-supply devices, smoke indicators, and so on. These aim at elimination of visible smoke.

As has been shown, however, visible smoke is only a comparatively small part of the problem. There are solids that do not color the smoke stream, and noxious and obnoxious gases. To combat the evil of solids, there are various types of separators. While admitting that such devices have great merit, the fact remains that, because of expense, their application is limited.

WHAT MUST BE DONE

There is already adequate engineering knowledge awaiting application, and at the proper time the engineer must introduce such combustion processes and power generating procedures as will solve as completely as possible the dust and smoke problem of the large cities. The great advances that have been made recently in the distribution of natural gas in the southern and mid-western states reveals the ability that the engineer has in solving his part of the problem when he has the raw material—fuel in this case—and support and encouragement.

In advance of the time when electricity, generated at central points or at the mine, will furnish power, heat and light, and so give us the *White City*, it is felt that the manufacture and distribution of steam by central stations for both heating and industrial processing use will also help to rid cities of smoke from small and often poorly operated plants. There are now over thirty companies in the steam business, mostly in the northern and central states.

It seems to the writer that the necessities for a hygienically clean atmosphere, and not one merely free from visible smoke, may be grouped about as follows:

1. Solids not to be discharged from stacks as at present. Similar elimination or neutralization of noxious and obnoxious gases. Once these impurities are in the air, they cannot be controlled.

2. Control of automobile exhausts, so that the present large volume of products of incomplete combustion from this source shall be elminated.

3. Such methods of street and roof cleaning, and care of railroad rights of way as will reduce to the minimum the amount of dirt that can be picked up and carried by the wind.

4. Reducing to the minimum the bare spaces from which dirt may be picked up.

This may seem Utopian, but consider what has been done for the water and food supplies. It is the engineer's job, and the engineer is capable of doing it.

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DISCUSSION

H. H. KIMBALL (WRITTEN): This paper presents much valuable information in a small space. Probably on account of its brevity one minor point is not quite clear.

Of the three well-defined dust layers referred to by Mr. Meller, all are resting on the surface. The upper limits of the first two are well known through frequent observation from mountain tops or from airplanes. The upper limit of the third is revealed through its optical effects. But the paper is dealing primarily with the contamination of the air in cities by dust of its own making, including smoke, which is most dense when wind, diurnal convection and the convective action of storms are all lacking. In consequence, dust that might be distributed through a layer seven miles thick, and over an area extending many times seven miles from its source, is confined to a layer in the immediate vicinity of its origin, and whose thickness is better expressed in feet than in miles.

This smoke cloud resembles the smudge with which growers of citrus fruits formerly thought it necessary to cover their orchards to protect them from frost. But the grocer, the tailor and the housewife complained of the serious damage to food and fabrics from the smudge. Then came the men of science to demonstrate that under conditions favoring frost a temperature inversion exists, which makes it more economical to warm the surface cold air layer by burning the fuel than to try to prevent its cooling by covering it with an objectionable smudge.

Similarly it has been demonstrated that the damage from smoke, and the waste of fuel in producing it, outweigh the cost of improved stoking methods that greatly reduce the smoke, or, in some cases, even the cost of installing in stacks dust precipitators that practically eliminate smoke. The public is greatly indebted to Mr. Meller and others working in his field for what has been accomplished in smoke abatement, but much remains to be done before our industrial cities can be considered clean. The goal will be reached only when an enlightened public demands it.

DR. C. B. MAITS (WRITTEN): Mr. Meller has called attention to the fact that the average adult inhales approximately 37½ lb of air per day. If this air carries solids and gases that are detrimental to any parts of the human system, it is obvious that, by the cumulative effect of such poisons, the healthy condition and the functioning of such parts may be impaired.

There is no question that air pollutants act as irritants to the eyes, nose, throat and lungs, and that some also enter the gastro-intestinal tract. The irritation by smoke particles has been considered as predisposing to acute diseases of the respiratory tract such as pneumonia. We know that pneumokoniosis, an affection of the lungs, develops in those who habitually inspire dust laden air.

Recent investigations in the United States, by C. E. Green in England and by Bertillon in France have shown that there exists a relation between air pollution and Cancer, death rates being much higher in coal burning than in wood burning areas. Green reports that, "the excessive presence of SO_2 gas from the use of coal with a high sulphur content was present with the highest incidence of Cancer." The tar released by the combustion of coal above England alone is estimated to be 500,000 tons yearly. It has been clearly shown by many research workers that Cancer can be and has been produced experimentally by the application of coal-tar to the skin of rabbits and mice.

The importance of effects of smoke upon meteorological conditions has been mentioned in the paper—particularly as it reduces or, in some cases, cuts out entirely the beneficial shorter wave-lengths, as it aids in the formation or prolongation of fogs, and as it affects the electrical potential. These in turn affect the human metabolism.

The energy loss to industry due to devitalizing, dark, humid days represents a money loss reaching into millions annually.

While it is true, as Mr. Meller has indicated, that the effects of smoke on health are not thoroughly understood, many physicians have made studies to determine that they are bad.

COL. ELLIOTT H. WHITLOCK (WRITTEN): The information presented is very complete and exact because Mr. Meller and his associates at the Mellon Institute have more data, which have been collected over a longer period of time under a continuing study of this problem than any other group in this country.

Enormous expenditures are being made on large size experimental apparatus in the endeavor to control or to completely eliminate deleterious products of combustion. It is by the experience gained in the failures as well as the successes attending these efforts that we can hope for progress along the general lines of atmospheric purification.

Smoke abatement is a long first step in the right direction toward a solution of this problem. Improving the combustion conditions helps to eliminate black smoke and reduce the total volume of gases from the stack because of the fewer pounds of coal consumed. In addition, this reduces the percentage of deleterious gases because the percentage of CO_2 gas has been reduced. This gas cannot be considered deleterious since it is one of the components of air. Other deleterious products of combustion also may be controlled.

Some of the more recent ordinances adopted have recognized that the responsibility for this work is up to the engineer and require the head of the department to qualify for active membership in the national society.

F. O. ANDEREG: Sulphurous gases sometimes condense in the chimney, attacking the water there, forming salts which continue to do their deadly work for long periods of time. These gases also put acid into rain water sometimes to a surprising extent all over the northern parts of this country. When this dilute acid solution, which falls in rain and in snow, has an opportunity to get into buildings it carries on its deadly disintegration. The acid itself, of course, has a bad action. It combines with lime, dissolves out the lime from the building material and does some damage, but that is not the worst of this situation, for the crux of this leaching action, usually calcium sulphate, continues its deadly disintegrating effects long after the acid has been neutralized.

This may sound far-fetched, but there are examples about us in this neighborhood, where considerable damage from disintegrating influence of this kind have occurred, as you will see by examining these pictures.

(Several slides were shown on the screen.)

PRESIDENT L. A. HARDING: These slides were illuminating and I am sorry that we don't have with us this morning certain architects and owners who usually lay all these troubles to the poor general contractor.

Are there any members here who desire to discuss this paper from the floor?

E. A. JONES: Just one question about the slides showing deleterious effect of SO_2. I don't think we have any answer to that or any suggestion of an answer. Abating the carbon isn't going to eliminate SO_2 if the fuel contains sulphur.

E. C. EVANS: Smoke bulletins (No. 1 to 13) issued by the Mellon Institute give definite analysis of atmospheric conditions.

H. B. MELLER: Since only two of these bulletins are available, it is planned to publish a book soon dealing with the various phases of the problem. This first volume may be followed by another of a more technical nature.

P. NICHOLLS: This occasion is noteworthy for being the first time to my knowledge that the Society has ever caused to be included in its papers one on smoke abatement. Actually the ordinances under which the smoke abatement departments of our towns and cities work limit their control to industrial establishments and buildings housing 6 or more apartments; therefore the whole field of smoke from dwellings of 5 apartments and down is excluded from the control of the smoke department.

Although the smoke from residences does not usually look as dense and black as that from industrial plants, yet it carries more tar and tar acids, is more harmful to goods at least, and causes more trouble to the housewife. All this smoke comes from appliances and from installations that are embraced in the activities that the Society claims as its responsibility. Is it not, therefore, the province and duty of the Society to have more papers on smoke and its abatement and to aid in promulgating ideals of engineering practice for the apparatus, and the operating of the apparatus, used in this field which is at present free of control?

W. H. CARRIER: The pertinent point has been raised that the abatement of the appearance of blackness in smoke will not abate the nuisance as far as the sulphur dioxide is concerned. I think that is true. In larger centers of population, where this difficulty is getting worse and worse, much is due to small homes, as Mr. Nicholls points out. The fogs of London are caused by its millions of chimneys. Our sulphur dioxide condition is not only caused by larger plants in office buildings but by furnaces of residences as well. As long as we are burning coal with a large sulphur content, this must persist whether we produce visible smoke or not. I believe that the line of improvement lies either in the use of desulphurized coke, if that were possible, or gas. Probably in the use of gas we have the greatest chance to remove not only the nuisance of visible smoke but also to reduce the sulphur content, because in the process of production of gas much of the sulphur is removed. Unfortunately, some sulphur is still present and makes the use of gas as a fuel very objectionable. However, with newer processes using a liquid treatment rather than a solid treatment in the removal of the sulphur content of gas, I believe it is possible to reduce it to such a point that it will no longer be seriously objectionable as a nuisance in air pollution.

If we have natural gas, we don't need to do it but if illuminating gas is used, we need to do it, and it can be done.

Possibly we will find some cheap method of pollution removal in power plants. It will be, of course, at an increased cost per kw, but I think it may be worth the toll.

C. H. BERRY: A large power plant in London has a complete gas-washing establishment attached to it for removing the sulphur gases from the flue gas, as well as the dust.

H. M. NOBIS: I believe from the experience in Cleveland that it would be wise for the Society's Council to consider another subdivision on air pollution and dust problems. The problem is a serious one. In Cleveland the politicians wanted to get the smoke problem under their control again and the *Cleveland Engineering Society* put in a protest with the helpful assistance of the *Woman's City Club*. I am frank to state that if the engineers had not had the assistance of the women in Cleveland about 8 years ago no progress would have been made in the elimination of smoke.

In Cleveland we are trying to persuade the Community Chest to set aside a certain sum for supervision of the smoke reduction, and public education. As soon as the people are educated to the problem, we will have less and less smoke. Architects will construct better homes with more insulation which will make it profitable for the people to burn gas.

DR. JENNINGS (Pittsburgh): As a botanist in this district for about 25 years I can't help but have noticed some interesting effects of smoke or gases upon the vegetation. A few years ago most of the delicate ferns in the nearby Conservatory were killed in September due to sulphur gases in the atmosphere. Zones of almost absolute desert lie adjacent to some of our larger industrial plants. A short distance from them increasingly larger amounts of vegetation can be seen clustering here and there in the little hollows where it gets some protection. Farther away the vegetation becomes more nearly normal.

Those instances have been increasing and in that zone between the industrial establishments and the zone where people want to raise plants there has been disagreement and much litigation which is still going on.

What does the damage? Since carbon dioxide is one of the main constituents of smoke, and also is an essential food for plant life, it would appear that more smoke will produce better flowers and better plants and more prosperity and all that, but that isn't the whole story.

Along with the smoke, as you have already heard, goes sulphur dioxide for one thing. Gases, like sulphur dioxide, entering through the pores mainly in the under surface of the plant's leaf tend to make the more concentrated solutions at the point where there is least water. The effect of that gas begins to show first in the areas between the nerves. In this district trees and shrubs are spotted with reddish-brown areas between the main nerves of the leaves. If those diseased areas become too large, the leaf and perhaps the whole plant suffers. The food supply made by that plant is going to be smaller. So if you can eliminate sulphur dioxide from the gas you are going to confer a great benefit upon the plant life of the community.

Leaves don't breathe in the sense that they inhale and exhale the air. Otherwise, just as in our nostrils, there would be a deposit of soot, etc., in those spaces. Air enters, carbon dioxide and oxygen enter through those minute pores but only at a comparatively slow rate and the pores do not become clogged to any great extent.

However, if the plant leaf is covered by a layer of soot which in this section, due to tarry materials in the atmosphere, will strongly adhere to the leaf, the sunlight will be cut off and due to the loss of such energy which is utilized for the manufacture of food, the plant would not be able to make so much sugar, starch and food material and its natural growth would be greatly retarded.

MR. JONES: I have had some experience with gas which is purified by the scrubbing process which indicates the tremendous amount of sulphur dioxide which is discharged by an ordinary coal-burning plant. Where gas is purified by the most modern means, including scrubbing, we can remove all of the sulphur products in the gas but the carbon bisulphide, which unfortunately at present cannot be removed commercially by any process, which will not also absorb the methane which is one of the principal constituents of gas. However, in these plants the gas supplied contains probably less than 10 per cent of the original amount of sulphur contained in the coal which was used to produce the gas.

Nevertheless, in gas-burning appliances, due to the presence of moisture, we have very marked corrosive action. Certainly we have far less sulphur in the products of combustion from that gas, but we have a condition which makes its presence manifest and that shows very forcibly the amount of sulphur that is escaping from appliances burning certain grades of coal.

PRESIDENT HARDING: The national advertising campaign of our tobacco companies would indicate that the pollution of the atmosphere by smoke is not such a bad thing. I believe 20,696 physicians have testified to its beneficial effects.

We have two distinguished visitors this morning and I am going to call upon them to say just a word. Dean Holbrook of the University of Pittsburgh.

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DEAN HOLBROOK: The talk on smoke pollution in the atmosphere brought me back to my days in the Bureau of Mines when we were interested, with a number of cities, in this problem.

The responsibility is not entirely with the engineers. Unless the engineer can have behind him, public opinion solidly and continuously, he cannot do effective work in smoke prevention.

PRESIDENT HARDING: If there are no further comments, will you close the discussion, Mr. Meller?

MR. MELLER: Most of the questions which have been raised have been answered with the exception of that concerning sulphur. If a city uses a coal which, as mined, averages 2 per cent sulphur, and if it is economically possible to reduce the sulphur content, by better preparation, to 1 per cent, the sulphur problem is cut in half. That is one step in urban air conditioning.

No. 892

SMOKE AND DUST ABATEMENT

By MELVIN D. ENGLE,[1] BOSTON, MASS.
NON-MEMBER

THE concentration of industrial activity around the large cities of the world has brought to the inhabitants of those densely populated areas new problems of many kinds. The problems of law and order, sanitation, water supply, transportation, and communication were some of the first to receive attention. After years of research and experimentation and the expenditure of vast sums of money, more or less satisfactory solutions have been found, but the ideal has not been achieved and probably will not be for some time.

Another problem resulting from the concentration of industry and habitation is that of air pollution. In 1661 strenuous objections were raised to the dust from sea-borne coal being unloaded in London, but according to the literature on the subject nothing was done to abate the nuisance. One writer referred to it as "the hellish cloud of sea-borne coal which maketh the city of London resemble the suburb of hell." During the period of almost three centuries since this blast against air pollution, the nuisance has increased many fold, but it has been only during the last decade that the problem has begun to receive the attention it deserves.

The main problem can be logically divided into three principal subdivisions; namely, smoke, dust, and noxious gases.

SMOKE

The dictionary defines smoke as "visible vapor or volatile matter that rises from a burning substance," but for the purpose of this discussion smoke will be defined as the finely divided solid matter formed by the incomplete or arrested combustion of the volatile hydrocarbons in fuel.

Prof. W. E. Gibbs in his book "Smoke and Clouds" classifies smoke as those particles ranging in diameter from 1/1000 to 1/10 of a micron. A micron is 1/1000 of a millimeter. This gives an idea of the infinitely small size of smoke particles.

It is furthermore known that smoke particles will not settle in still air.

[1] Station Engineering Department, The Edison Electric Illuminating Company of Boston.
Presented at the 37th Annual Meeting of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, Pittsburgh, Pa., January, 1931.

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Sometimes several smoke particles will stick together and fall to the earth, but as long as the particles remain separated, they will not settle. This means

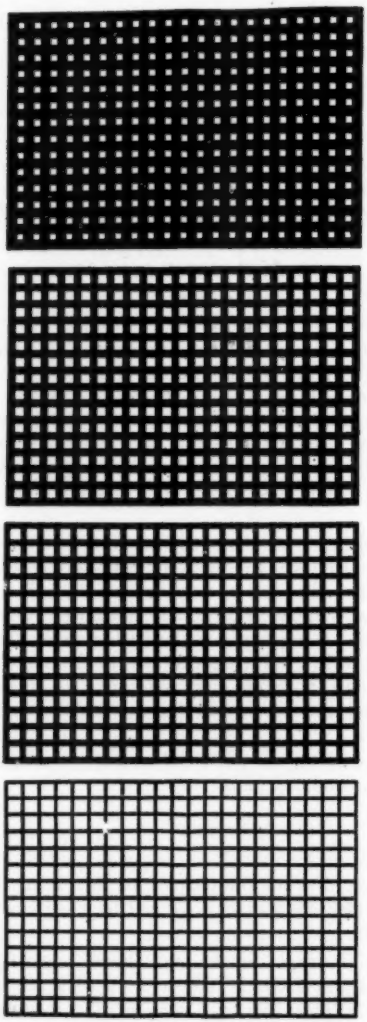


FIG. 1. RINGELMANN'S SCALE FOR GRADING THE DENSITY OF SMOKE. LEFT TO RIGHT: 1. EQUIVALENT TO 20 PER CENT BLACK; 2. EQUIVALENT TO 40 PER CENT BLACK; 3. EQUIVALENT TO 60 PER CENT BLACK; 4. EQUIVALENT TO 80 PER CENT BLACK

that the problem of catching them is almost hopeless in the light of present-day knowledge.

It has been found, however, that it is possible by the proper design and operation of the fuel burning equipment and furnaces to practically prevent

the formation of smoke. It may be definitely stated, therefore, that the problem of smoke abatement can be solved by the selection of the proper fuel burning equipment and furnace design for the particular fuel to be burned, and by the proper operation of this equipment.

Where the equipment and furnaces are already installed, the problem of smoke prevention is not so simple. A change in the fuel burned or any one of a number of things can be done to improve conditions. The finding of the proper solution for a particular installation is quite often enough to tax the ingenuity of the cleverest engineer.

The first step taken toward air pollution abatement was the passing of the so-called *smoke laws.* Such laws were on the statute books for several years before any standard means were available for measuring smoke density. The well-known Ringelmann charts, shown by Fig. 1, were then devised and are the accepted measure of smoke density today.

The smoke laws have failed materially to abate the nuisance for two reasons: *First,* they were not based on a complete understanding of the problem; and *second,* because such nuisances cannot be abated by the passing of laws alone. They require extensive research and the expenditure of vast sums of money for experimentation and the development of the necessary equipment.

The experience of the city of Pittsburgh well illustrates the point. The Mellon Institute made a soot fall test for the city in 1914 and a smoke law was passed. In 1924 the same institution made a second soot fall test in exactly the same manner, and it was found that although the tarry matter in the soot collected was reduced from 10 tons to 3½ tons per square mile per year, the total solids had increased from 1,000 tons to 1,400 tons per square mile per year. The smoke nuisance had been abated, but the air pollution nuisance had increased 40 per cent.

DUST

For the purpose of this discussion, dust may be considered as any solid particles larger than the largest smoke particle, or any particle larger than ⅒ of a micron.

Dust particles varying in sizes from ⅒ to 10 microns in diameter settle in still air at a constant velocity depending upon their size, and particles larger than 10 microns settle in still air at a constantly increasing velocity.

It is obvious that the problem of separating dust from boiler flue gases becomes increasingly difficult as the size of the dust particles grows smaller. A relatively simple device will collect the large particles of coke or ash that are occasionally discharged with the flue gases, but the problem of collecting the finer particles of dust is extremely difficult. Unfortunately, by far the greater part of the dust in boiler flue gases is extremely fine.

The exact size of dust particles in the flue gas of coal-fired boilers varies greatly depending upon a number of factors, but for pulverized coal-fired boiler flue gas, in general, 90 to 95 per cent will pass through a 200-mesh sieve, and from 65 to 80 per cent will pass through a 325-mesh sieve. Particles that will just pass through a 325-mesh are approximately 43 microns in diameter. No reliable data are available on the size of the dust in the flue gases from stoker-fired boilers, but it is known, in general, that by far the greater part of it will pass through a 200-mesh sieve.

XUM

The ideal, of course, is to catch all dust particles and at the present stage of development of the art of dust catching it might as well be admitted that it is not known how to accomplish the ideal. However, practically all of the coarse particles and a fair percentage of the smaller particles can be caught. As the art of dust catching progresses, and as the equipment is improved and new types of equipment are developed, it will be possible to catch a larger and larger percentage of the finer dust particles.

Fortunately, the larger dust particles constitute the greatest nuisance and as

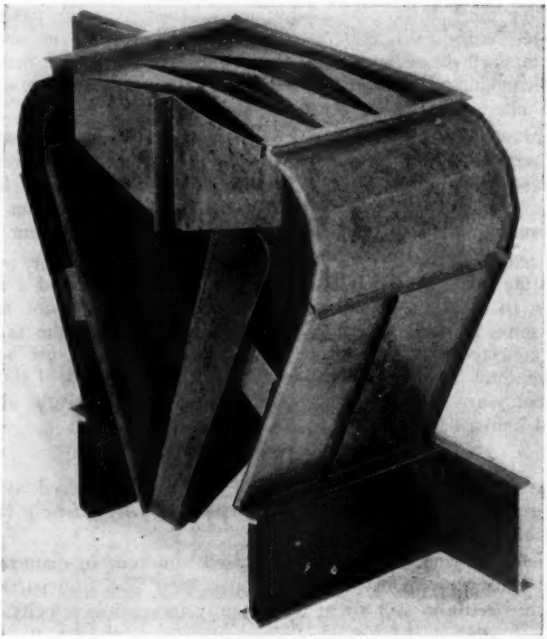


FIG. 2. DUST CATCHER

the size of the dust particle diminishes, the seriousness of the nuisance diminishes also. Some engineers will question the accuracy of this statement, but a careful study will prove its accuracy. In support, it is known that fly ash which will pass through a 300-mesh sieve but which will be retained on a 325-mesh sieve will carry approximately four miles when discharged from a chimney 200 ft. above the ground into a 10-mph wind. Particles of fly ash one-half this size will carry approximately 20 miles when discharged to the atmosphere at the same height above the ground in a wind of the same velocity. Consequently, for a given quantity of dust discharged to the atmosphere, the finer the dust, the smaller will be the deposit in tons per square mile per year, and the dust load per cubic foot of atmosphere. This statement is offered

as a basis of discussion with the belief that such discussion will be helpful in the solution of the problem.

The measurement of the quantity of dust existing in the air is an extremely difficult one, and no two methods in use today will give the same answer. The impact dust counters, in which a measured quantity of air is drawn through an aperture and impinged against a microscope slide, give the highest dust counts of any methods in general use.

Doctor Free has conducted some very interesting dust counts in the city of New York, and he reports counts varying from 35,000 to 880,000 particles per cubic foot of air. These counts were taken from the seventeenth story of a building at 23rd Street and Fifth Avenue, New York City, and he estimates that the dust counts at the street level would be 50 to 100 per cent greater.

By measuring the size of the dust particles in the air, Doctor Free has estimated that there is an average of about one ounce of dust per 10,000,000 cu. ft. of air and that the air over the entire city of New York contains approximately four or five tons of dust. Particular attention is called to this figure, published by a recognized authority, to offset the statements that have been made by others that the air over the city of New York contains several thousand tons of dust.

Where does this dust come from? Doctor Free reports that his examination of the dust in the air above New York City shows that over 90 per cent of it is "clearly ordinary soot, that is, unburned carbon." Particles of rubber from automobile tires, asphalt, brick, rock, organic materials, and of leather, cotton fibers, hair, and other materials too numerous to mention contribute to the dust. The absence of ash particles in the air is particularly noteworthy, but it is not surprising in view of the large amount of carbon that must be discharged at low levels by the exhaust of automobiles, and the chimneys of residences, small apartment houses, office buildings, and small industrials, burning large quantities of coal in completely water-cooled furnaces and at very low furnace temperatures. Large furnaces operating at high temperatures and under competent supervision do not discharge as much carbon to the atmosphere as is generally supposed.

The first real start in the cleansing of the flue gases discharged from chimneys was made in those industries where the value of the dust reclaimed would pay for the necessary equipment for collecting. Metallurgical and cement plants have struggled with the problem for years.

The dust reclaimed from the flue gases of pulverized coal-fired boiler plants is valueless today. Those who have been willing to spend money to abate the nuisance of air pollution by these plants without a chance of a return on the money invested in the collecting equipment deserve special commendation for the public-spirited attitude which prompted them.

The companies operating large boiler plants are fully alive to the necessity for smoke and dust abatement. They have gone a long way toward the elimination of smoke, and they are demanding equipment for the catching of the dust in the boiler flue gases. These demands have resulted in the development of a large number of different types of dust catchers, and many more are in the process of development.

XUM

The dust catchers available may be divided into the following general classes:

1. Settling Chambers.
2. Dust Traps.
3. Centrifugal Separators.
4. Electrostatic Precipitators.
5. Gas Scrubbers.
6. Filters.

Settling Chambers

Probably the earliest attempts to catch dust in large quantities were made by the operators of blast furnaces. The gases were discharged into large settling chambers and the velocity of the gases reduced to a point where a considerable portion of the dust settled out by gravity. The bottoms of these chambers were provided with dump plates through which the collected dust was removed. These settling chambers were very effective for removing the large particles of suspended matter and went a long way toward reducing the air pollution nuisance due to the operation of blast furnaces. They were unable, however, to effectively remove the fine dust, and where large quantities of gas are to be handled these chambers become so large and expensive that they are not commercial. Furthermore, it is not practical to install such chambers on the suction side of induced draft fans operating at high suctions. They marked the earliest attempts to solve the problem of air pollution by dust, however, and played an important part in the development of the art of dust catching.

Dust Traps

It is not readily determined whether the so-called dust traps or the centrifugal dust catchers were the next step in the development, but probably the dust trap came next. Operators of settling chambers probably found that by varying the shape of the chambers and by the use of baffles higher efficiencies of collection could be obtained, and these experiments probably led to the development of the earliest forms of dust traps. Many different types are manufactured today, and the designs show plainly the ingenuity of those who have developed them. The results obtained in some cases are almost startling.

It would appear that dust traps should be able to operate equally well at any gas temperature. Recent experience has shown, however, that the temperature of the gas does have a marked effect upon the efficiency of some designs. The following descriptions of several different designs are offered without reference to which was developed first.

Fig. 2 shows a dust catcher designed for installation in vertical ducts or over boiler outlets. As can be seen from the illustration, the gases are subjected to an abrupt change in direction and the dust is deposited in the center chamber from which it can be removed by gravity by means of spouts.

Another company offers the cinder trap shown by Fig. 3. This equipment divides the gases conveying the cinders into thin vertical sections and subjects these vertical strata to several abrupt changes in direction, as illustrated. The cinders are caught in the heart-shaped elements and are conveyed downward

by gravity and a small stream of the gas to the hopper below. The cinders are deposited in the special-shaped hopper and the small quantity of conveying gas is released by this hopper into the main gas passage. By splitting up the gas into the narrow strata, the particles of dust are only required to pass through a very thin strata of gas to the collecting elements. These traps have proved very effective for cinders of relatively large size. The manufacturer offers this type especially for stoker-fired installations.

The same company has under development a dust trap for pulverized fuel fly ash that seems to have possibilities. An illustration is not available, but in

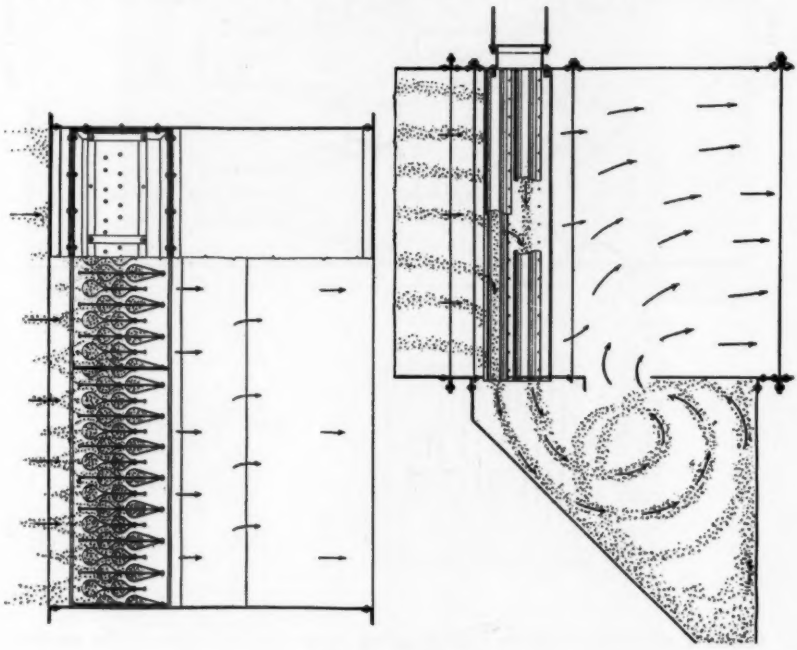
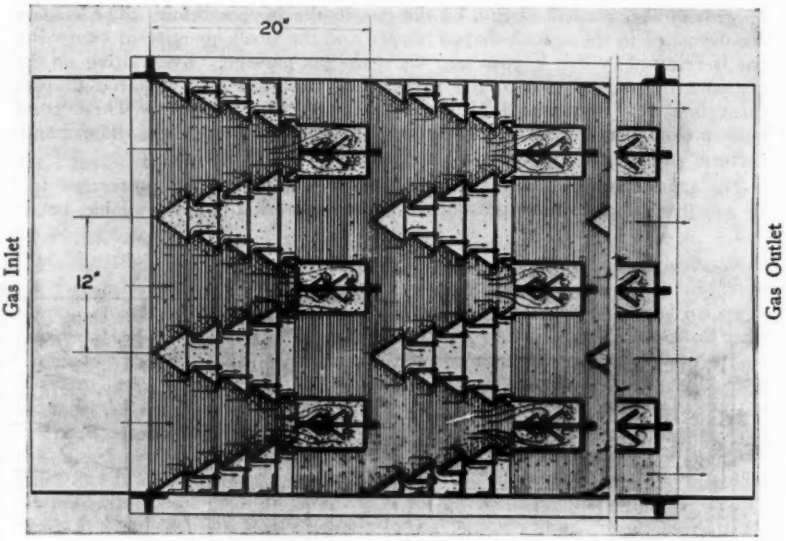


FIG. 3. CINDER TRAP

general the gas is split up into thin vertical strata by a series of barbed and corrugated plates suspended over dust hoppers. The corrugated plates form alternate nozzles and expansion chambers. In the passage through the apparatus, the dust is blasted against the barbed surfaces to which it adheres. The barbed plates are set in groups forming sections approximately two feet wide. At intervals the sections are closed off by automatic dampers and the dust is shaken off by automatic rappers into the hopper below. This trap is under development, and further details are not available at this time.

A New York company manufactures the dust trap shown by Fig. 4. They claim to concentrate and separate the dust by creating free spiral vorticose. In-



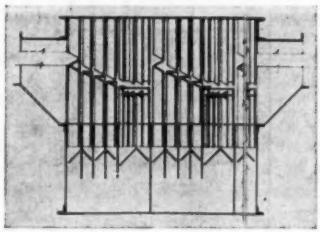


FIG. 4. CINDER CATCHER (Vorticose Type)

verted V elements as shown are supported above a dust hopper. The number of elements in the path of the gas is varied from three to seven, depending upon the fineness of the dust to be collected, and the number of units in width and height is varied to accommodate the quantity of gas to be handled. A small quantity of gas flows downward into the hopper as a conveying medium and automatic rappers help to dislodge the dust which collects on the plates. The hopper is maintained in a quiescent state, and the dust is settled out from the small quantity of gas which enters the hopper as a conveying medium.

Another New York company manufactures barbed-plate type dust collectors.

Fig. 5 shows how the barbed and corrugated plates are sometimes arranged on a horizontal shaft. As the shaft is slowly rotated, certain portions of the plates are always in a quiet chamber while the gas flows through the other plates on the shaft. The plates in the quiet portion are vibrated and the

dust removed. For large gas quantities, the elements are suspended above dust hoppers in a steel casing, and the sections are closed off for rapping by means

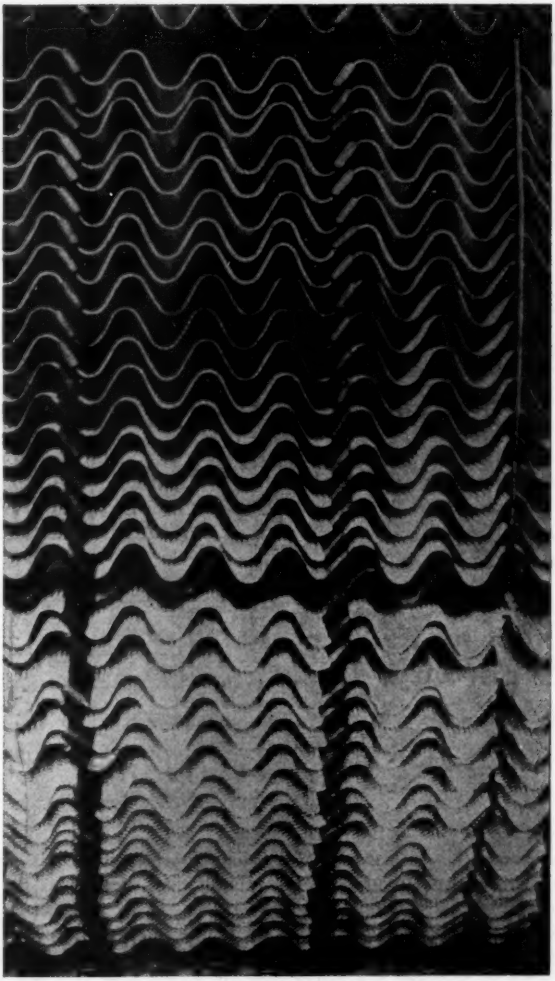


FIG. 5. BARBED-PLATE CATCHERS

of dampers. The machines are built on the unit principle, and the size and number of units is varied to meet the conditions of a particular installation.

A number of attempts have been made to design pulverized fuel-fired furnaces

in such a way as to catch a larger percentage of the dust than is normally caught. Nothing has been developed that would indicate that such a design would make it unnecessary to install other apparatus, but some such design might well catch the larger particles of ash. Most of those tried so far have wet bottoms where the ash is maintained in a molten state and tapped off the same as blast furnace slag.

Centrifugal Catchers

Many types of centrifugal collectors have been used with considerable success. Fig. 6 shows the ordinary *garden variety* of cyclone separator which is

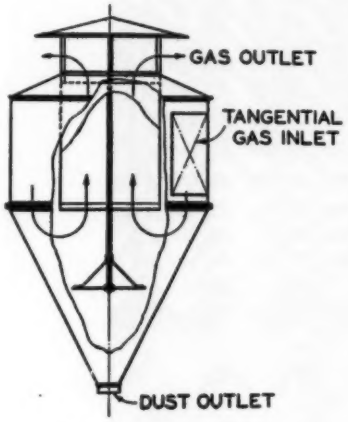


FIG. 6. SIMPLE CYCLONE

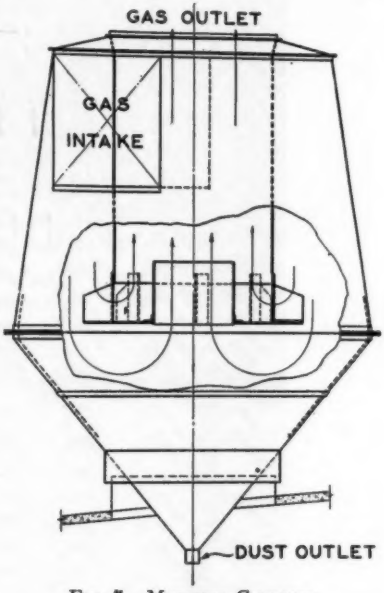


FIG. 7. MODIFIED CYCLONE

used so extensively for separating solid matter of many kinds from air or gas. All of these centrifugal collectors obtain dust separation by projecting the particles tangentially out of the gas stream. The effectiveness of these collectors varies directly as the specific weight of the dust, and as the square of the tangential velocity, and inversely as the radius of rotation. The type of collector shown is effective for heavy dust of appreciable size but has proved entirely inadequate for such dust as carried in the flue gases from coal-fired furnaces.

Fig. 7 shows a modified form of cyclone catcher that will catch a large amount of cinders from stoker-fired boilers, but fails to catch the extremely fine dust. The dust is concentrated in the outer layer of gas, and the inner

layer is allowed to escape to the outlet, as indicated by the arrows. The outer layer of gas in which the dust has been concentrated continues to the bottom of the collector and is discharged through the bottom of the outlet tube. The straightening vanes shown in the main gas outlet reduce the spin in the outlet pipe and thereby reduce the pressure loss through the apparatus.

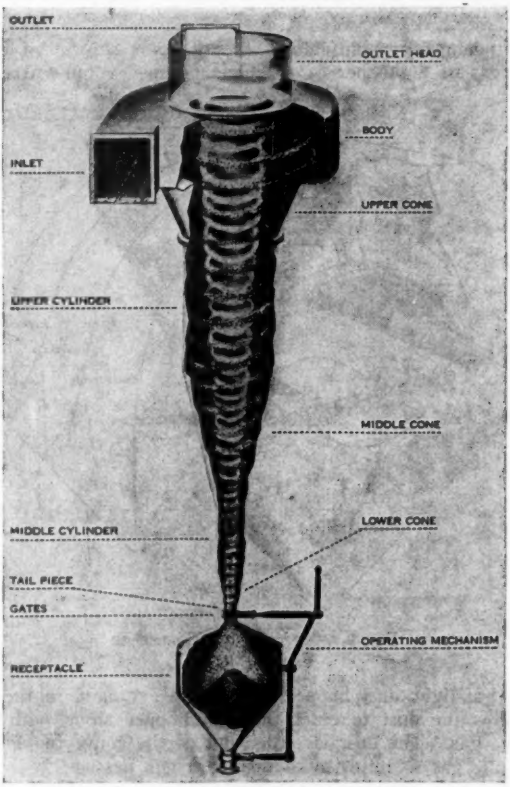


FIG. 8. MODIFIED CYCLONE CATCHER

Fig. 8 shows another modified form of cyclone collector. This collector consists of a main casing of volute shape through which the dust enters tangentially. Beneath the volute, a series of alternating truncated cones and cylinders are arranged terminating in a dust outlet at the bottom. The gas is discharged through a central opening in the top of the main casing or through this opening into a scroll-shaped outlet head. The manufacturer claims that the design creates a cyclone action without internal turbulence.

The dust-laden gas entering the collector tangentially is forced by its velocity to take a path following the shape of the volute. The dust particles are forced outward against the periphery. In addition to moving centrifugally outward, the dust is forced downward and leaves the volute in a downward direction following a spiral path which is directed and increased in velocity by the alternating truncated cones and cylinders to the dust outlet at the bottom. The gas forms a vortex in the center of the collector and passes upward through the outlet.

An English company manufactures the collector shown by Fig. 9. This collector consists of a number of triangular shaped volutes arranged side by

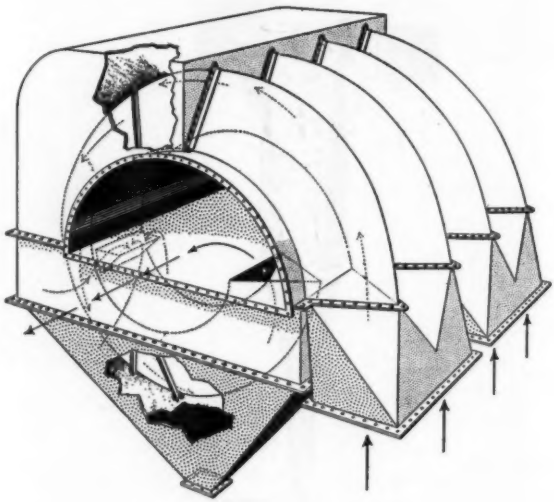


FIG. 9. CENTRIFUGAL COLLECTOR

side above a gas-tight dust hopper. Slots are arranged in the apex of the volutes to allow the dust to enter the dust hopper along with a small percentage of the gas. The dust is settled out of the conveying gas in the dust hoppers and the gas is returned to the main gas passage.

Another centrifugal collector is shown by Fig. 10. As can be seen, this collector consists of two horizontal concentric cylinders. The gas is admitted tangentially through the outer cylinder at one end into the space between the two concentric cylinders, and leaves through the center of the inner cylinder at the opposite end. In this manner the gas is given a spin, and the dust is thrown out through slots in the outer cylinder into a suitable gas-tight hopper.

A number of attempts have been made to make the induced draft fan act as a dust separator. Fig. 11 shows one of these designs. The cut clearly shows the separating action that takes place. The gas containing the cinders enters the inlet of a paddle-wheel fan, the blades of which are provided with chan-

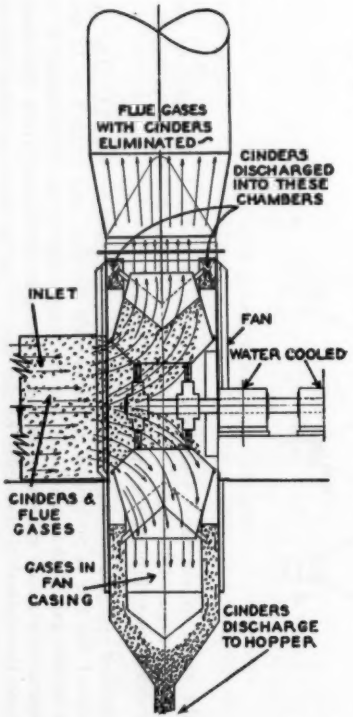
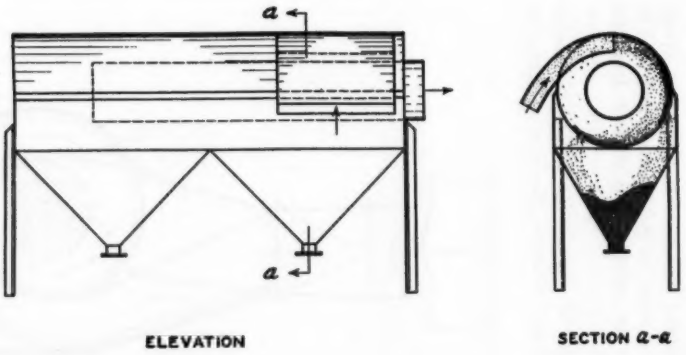


FIG. 11. CINDER-CATCHING FAN

FIG. 10. CYLINDRICAL COLLECTOR



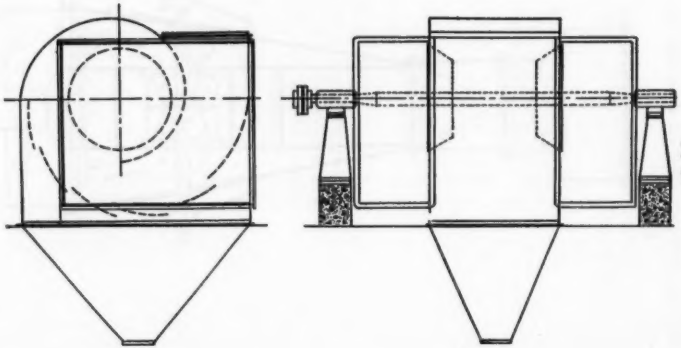


FIG. 12. CYLINDER-ELIMINATING FAN

nels to catch and direct the cinders into the chambers in the sides of the casing. This fan makes use of the sudden changes in gas direction and velocity that occur in all paddle-wheel type fans to separate the cinders from the gases.

A cinder-eliminating fan is shown by Fig. 12. In this fan advantage is taken of the concentration of dust due to centrifugal action in the layer of gas next

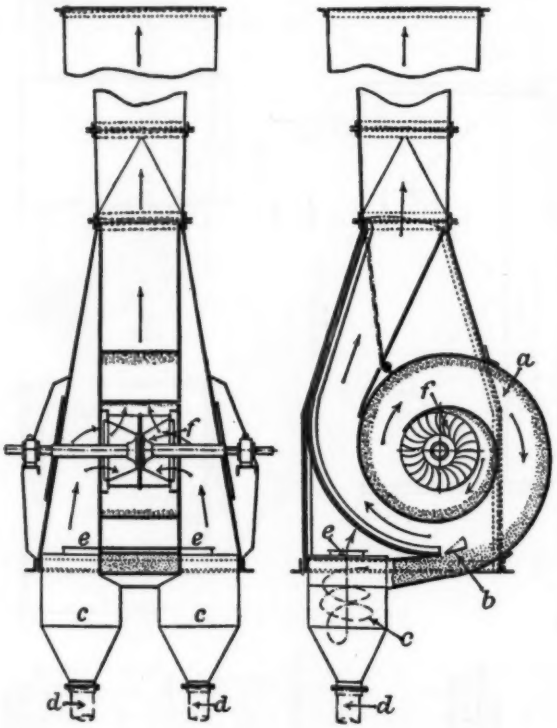


FIG. 13. CINDER-COLLECTING FAN

to the scroll of the fan. Openings are provided in the bottom part of the scroll and a gas-tight dust hopper is arranged below the fan to collect the dust discharged through the openings in the scroll.

Another cinder collecting induced draft fan is shown by Fig. 13. The involute of the fan is prolonged to concentrate the dust and from 5 to 15 per cent of the gas is drawn off and the dust separated from it by means of a comparatively small cyclone collector. The gas discharged from the small cyclone is returned to the inlet of the fan. One or more cyclones are in-

stalled, depending upon the dust collecting efficiency desired and the character of the dust to be separated.

All of these centrifugal devices have the serious problem of erosion to overcome, and the space requirement is often embarrassing. The advantages of this type of equipment are obvious.

Electrostatic Precipitators

In 1886, Sir Oliver Lodge in a lecture before the Liverpool Section of the *Society of Chemical Industry* pointed out the possibility of separating dust

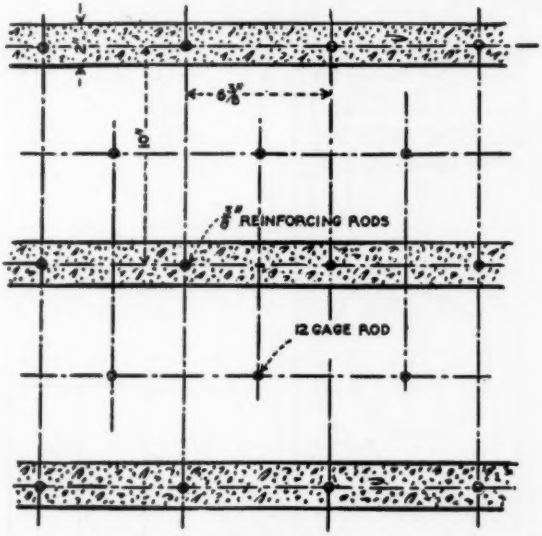


FIG. 14A. PLAN SHOWING DETAILS OF ELECTROSTATIC PRECIPITATOR CONSTRUCTION

and vapor from gases by electrical means. In 1903-1904, Dr. F. G. Cottrell became interested in the Lodge experiments and applied the principles to the precipitation of sulphuric acid mist. After years of experimentation, he produced a successful electrostatic precipitator. Electrostatic precipitators have been installed in a number of plants in the United States for catching the fly ash in the gases from pulverized coal-fired furnaces.

Figs. 14a, 14b and 14c show the design of and efficiency curve for the electrostatic precipitators installed at the Trenton Channel Plant of the Detroit Edison Company. The present design used for the precipitation of fly ash from the gases of pulverized fuel-fired boilers was developed at that plant.

The precipitators consist of gas-tight chambers in which are suspended vertically, concrete slab collecting electrodes parallel with each other, as shown in this figure. The concrete electrodes are two inches thick and have imbedded

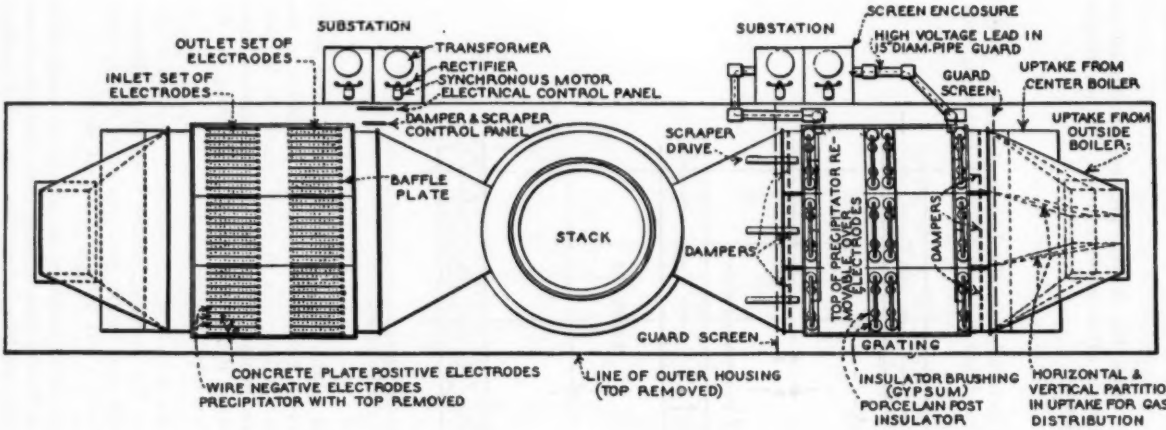


FIG. 14B. (ABOVE) PLAN SHOWING RELATION OF ELECTRICAL PRECIPITATORS, ELECTRICAL EQUIPMENT, STACKS, ETC.

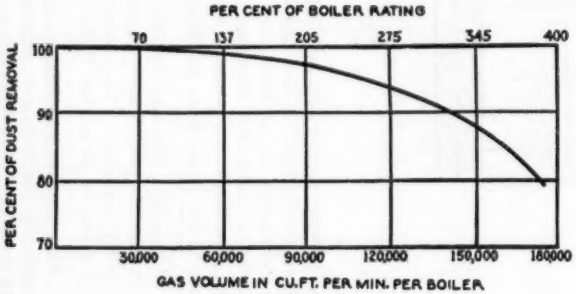


FIG. 14C. (LEFT) TYPICAL EFFICIENCY CURVE FOR ELECTROSTATIC PRECIPITATOR

in them steel reinforcing rods which act as the positive electrodes. The plates are spaced 10 in. apart, center to center, and in the space between are suspended bare rods as negative electrodes. Alternating current at 75,000 volts is rectified by a mechanical rectifier to supply unidirectional current to the electrodes of the precipitators.

This high potential ionizes the dust-laden gas as it passes through the precipitator and causes the dust particles to take on a high potential charge. An

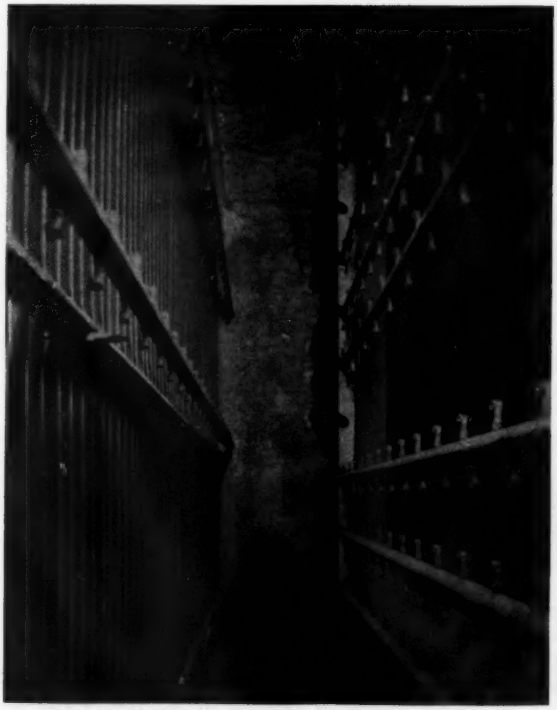


FIG. 15. SIMPLE GAS WASHER

electrically charged particle under these conditions tends to move toward the negative electrodes or the concrete slabs in these precipitators. The dust clings to the concrete slabs until it builds up to a thickness that will cause it to drop off due to its own weight into the dust hoppers provided below. The precipitators are divided into sections and each section is provided with suitable dampers so that sections can be closed off and the dust scraped off the concrete slabs, if necessary, while the other sections are in operation. The efficiency of this type of dust catcher varies with the dust loading, size of the dust particles, and the time that the dust-laden gas remains in the catcher.

Gas Scrubbers

The dust in boiler flue gas has been successfully removed by means of gas scrubbers. Fig. 15 shows the inside of a gas washer installed for removing the dust from the gases of a pulverized coal-fired boiler in the Calumet Station of the Commonwealth Edison Company of Chicago. This scrubber consists of one set of sprays and a set of eliminator plates with flooding nozzles for washing the dust from the eliminator plates.

The big problem in developing a satisfactory wet dust catcher is to find materials that will resist the corrosive action of the wash water. The water

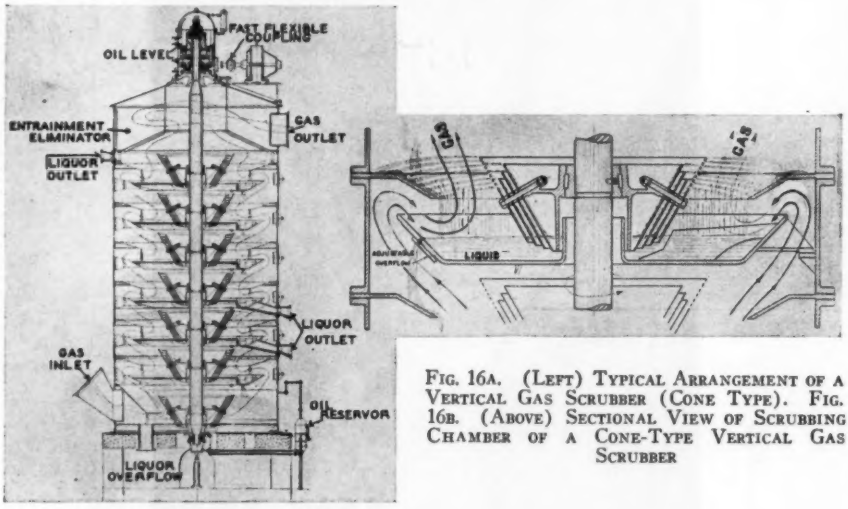


FIG. 16A. (LEFT) TYPICAL ARRANGEMENT OF A VERTICAL GAS SCRUBBER (CONE TYPE). FIG. 16B. (ABOVE) SECTIONAL VIEW OF SCRUBBING CHAMBER OF A CONE-TYPE VERTICAL GAS SCRUBBER

from these scrubbers contains sulphuric and sulphurous acids and very few suitable materials are available that will resist the action of the latter for a reasonable length of time.

A number of European plants have installed sprays in the chimneys of the boiler plants and report successful removal of the dust. The problem here would seem to be to prevent discharging acid water particles from the top of the chimney. A very low gas velocity will convey the smaller water particles and it is necessary that this possibility be eliminated before this scheme can be adopted generally.

Fig. 16 shows a vertical type of scrubber that has been used for blast furnace gas. As can be seen, the scrubber consists of a vertical shell with a gas inlet at the bottom and a gas outlet at the top. Inside the shell are a number of baffles and troughs which cause the gas to flow through a number of water sprays. The water sprays are provided by a rotating, multiple cone water atomizer and also by the streams of water dropping from the upper baffles to the lower.

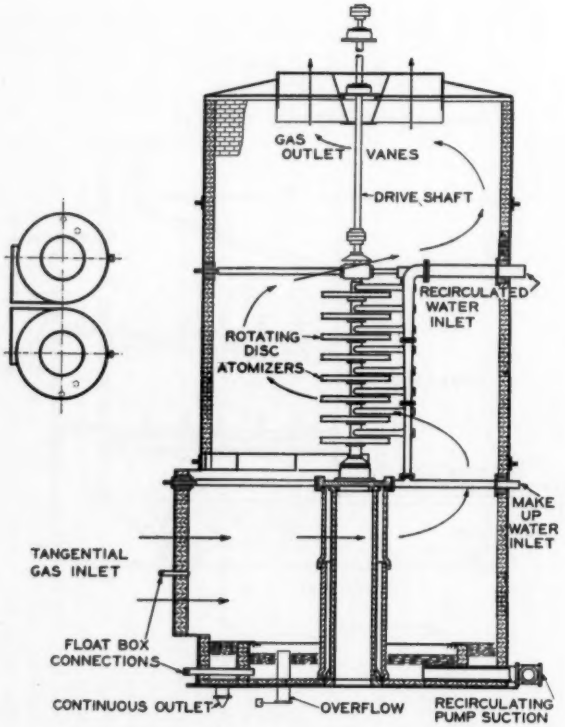


FIG. 17. VERTICAL GAS SCRUBBER (DISC TYPE)

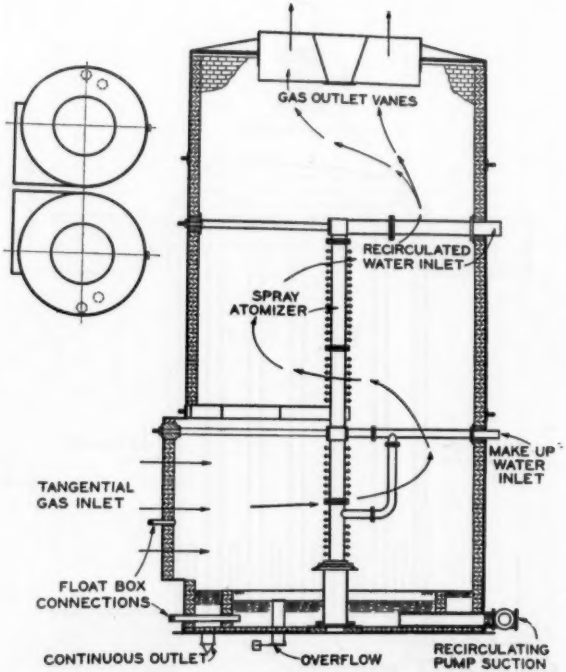


FIG. 18. VERTICAL GAS SCRUBBER (SPRAY TYPE)

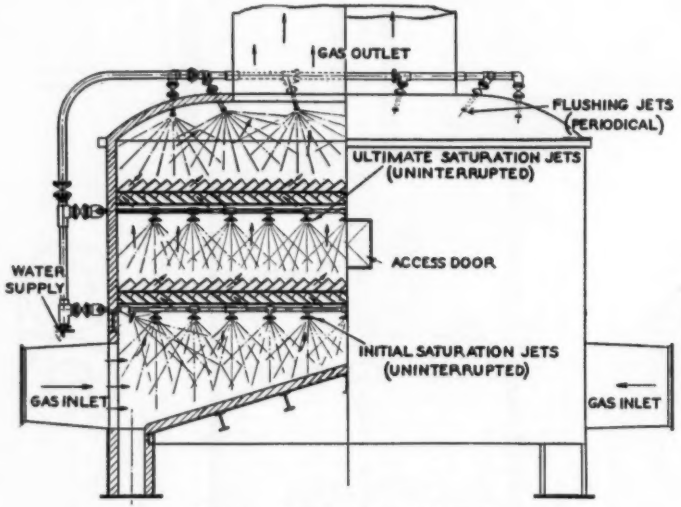


FIG. 19. ENGLISH GAS WASHER

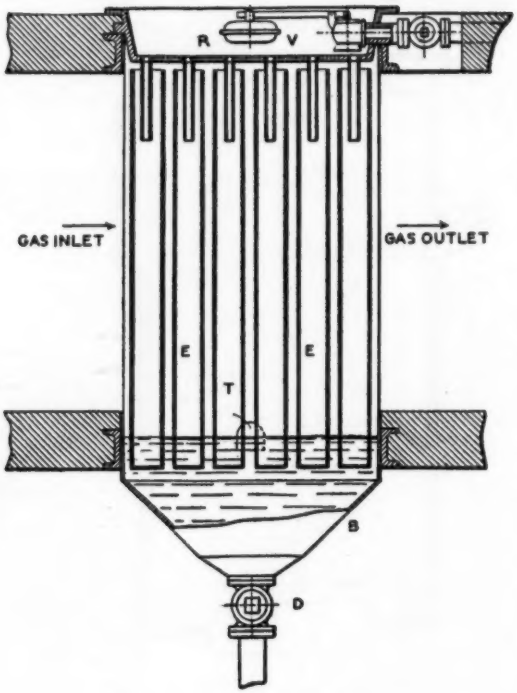


FIG. 20. FRENCH GAS SCRUBBER

Fig. 17 and Fig. 18 show the two types of vertical scrubbers installed in the new Kneeland Street steam heating boiler plant of The Edison Electric Illuminating Company of Boston, Massachusetts. Fig. 17 shows the design that utilizes the revolving disc atomizer. The gas is admitted tangentially at the bottom so as to give the gas a spin and is released at the top through straightening vanes which iron out the spin of the gas and reduce the pressure drop through the scrubber.

The bottom of the scrubber forms a tank for the collection of the dust and water. A certain quantity of water is drawn off continuously to remove the dust and the remainder of the water is recirculated as the scrubbing medium. By spinning the gas with the tangential inlet the larger particles of dust are thrown out more readily by centrifugal action, and the water particles are thrown through the gas stream. Furthermore, the gas spin throws out all water particles and discharges moisture-free saturated gas from the scrubber.

Fig. 18 shows the design that utilizes the spray head type water atomizers. One boiler in the Kneeland Street plant is provided with the scrubbers equipped with the revolving disc atomizers, and the other boiler is provided with scrubbers equipped with the spray head type atomizers.

The bottom or tank portion of each scrubber is lined with lead and the bottoms and shells are protected with a lining of acid-proof brick laid in acid-proof mortar. The discs, spray heads, and struts are constructed of special acid-resisting metal. Other parts are of cast iron painted inside with an acid-proof paint and protected on the outside with an acid-resisting plastic material.

The rotating disc type of water atomizers undoubtedly give a finer spray than the spray heads. On the other hand, they are more expensive to install and maintain. The Kneeland Street plant installation in Boston should tell whether the more expensive atomizing device is necessary.

Fig. 19 shows an English design of gas scrubber. As can be seen from the cut, the gas is admitted at the bottom and released at the top of a cylindrical shell. Inside the shell are arranged two sets of sprays which are operated continuously, and two sets of baffles. Above the top set of baffles is installed a set of flushing sprays for periodically flushing off the baffles. The water is drained away at the bottom.

Fig. 20 shows a gas scrubber which differs radically from the types of scrubbers described previously. In this scrubber the shell is rectangular and the gas flow is horizontal. Inside the shell are arranged large numbers of closely spaced pipes, rectangular in shape, which are closed at the bottom and open at the top. Water is admitted through calibrated orifices into the top of the vertical pipes and overflows and keeps the outside of the scrubber pipes wet with a film of water. The gas is split up into thin vertical sections as it passes through the scrubber, and these vertical strata are subjected to several abrupt changes in direction. The dust is impinged against the wet scrubber pipes where it is caught and carried away to the sump below which removes the water and dust. By splitting the gas into the narrow strata, the particles of dust are only required to pass through a thin strata of gas to the collecting elements, the same as in the case of some of the dust traps. The water prevents

the re-entrainment of the dust particles, and the water that is evaporated increases the relative humidity of the gas and aids in the separating action.

Filters

Filters of various kinds have also been tried with considerable success. Fig. 21 shows a type of fabric filter. The cut shows a very small filter, but the same design is applied in filtering the dust from boiler flue gases. The dust-laden gas is admitted at the bottom of the gas-tight casing and is filtered through bag-type fabric filters of cotton, wool, or asbestos cloth, depending upon the temperature of the gas to be filtered. The filters are made in units

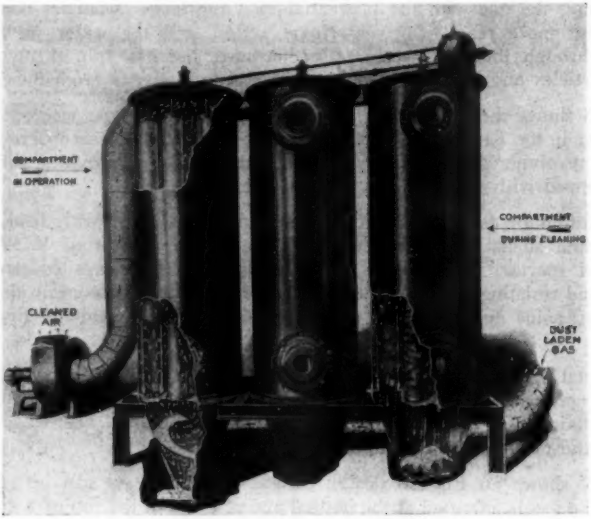


FIG. 21. FABRIC FILTER

so that individual units can be shut off with dampers and the bags shaken to remove the dust collected on the bags. In Fig. 21, the unit at the left is in operation while the unit at the right is being cleaned.

Fig. 22 shows a gravel filter now in the process of development. In this filter, gravel is used as the filter medium and is removed continuously at the bottom, screened to remove the dust, and returned to the filter at the top.

Practically all of the dust catching devices described are in a state of development, and it will probably be many years before the most satisfactory types are determined. It should be noted, however, that only the wet scrubbers make any attempt to catch the noxious gases.

The development of these and many other types of dust catchers clearly shows the general interest in the subject and the tremendous amount of research and development work that has already been done. Undoubtedly, within a few years, satisfactory and commercial equipment will be made available.

One of the embarrassing features of the subject is that after a dust catcher has been installed it is extremely difficult to tell exactly how thoroughly it is doing its work. The methods of test are many and each will give a different answer. The manner of conducting the test can vary the answer to suit the convenience of those responsible for the conduct of the test. There is a serious need for a standardized test code for dust collectors. Even though it

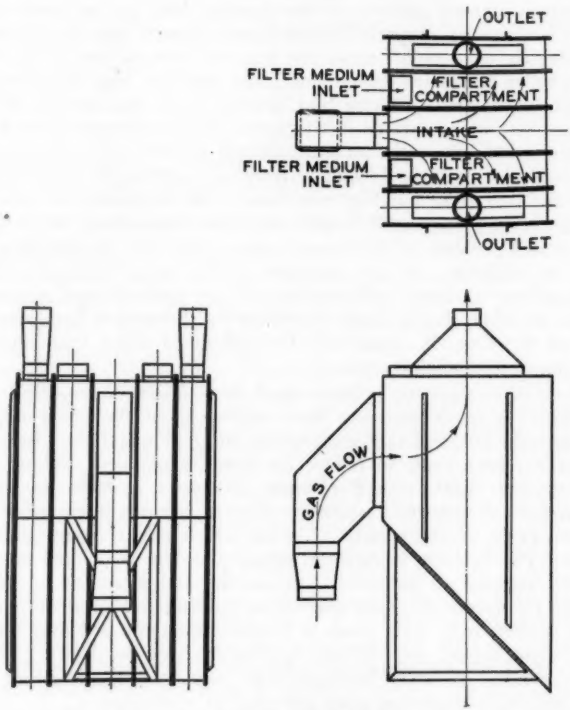


FIG. 22. GRAVEL FILTER

may not be perfect, if it will only give comparable results for the various tests in various plants, it will help immensely.

A very simple test would be advisable at this stage of developments, so that a large amount of comparable test data will become available in the shortest time possible. A code calling for a standardized test by means of standardized fabric filters would be useful, and would result in the collection of data that would be extremely valuable in the development of the art of dust catching. It is to be hoped that the formulation of such a code will be undertaken in the near future.

NOXIOUS GASES

The third subdivision of the subject is the pollution of the air by noxious gases which are harmful to animal and vegetable life and to building materials.

There is no question that the discharge of active gases of this nature to the atmosphere must be stopped. Those industries which have to deal with poisonous gases have already had to take steps to neutralize the gases discharged. It is not helping the problem, however, to believe the articles published which paint extremely gloomy pictures of the damage done by the gases discharged from the chimneys of properly designed and properly operated boiler plants. Disregarding the question of smoke and dust, the only noxious gases discharged from such boiler plants are the sulphur gases resulting from the combustion of the sulphur in the fuel. Where high-sulphur coals are burned, the problem may become serious enough to justify going to some length to neutralize the sulphur gases, but where the sulphur content of the fuel is less than 2 per cent, it would be extremely difficult to trace any appreciable damage to human and vegetable life or to building materials to the discharge of such gases to the atmosphere at a reasonable height above the surrounding buildings.

Where a real problem of this nature exists, however, a satisfactory answer seems to be available. A gas scrubber of the proper design, utilizing the proper scrubbing medium, will remove a large part of such noxious gases. If the gas scrubber should finally prove to be the proper equipment for the catching of the dust, it should not be difficult to make that scrubber also serve to absorb the sulphur fumes.

The effects of atmospheric pollution have been classified as medical, botanical, and physical. A great deal has been written about the effect upon human health, vegetable life, and the deterioration of inert materials. In some cases an attempt has been made to reduce the harmful effect to dollars and cents. Such figures are absurd and it certainly is unwise to build up hypothetical figures without a reasonable basis in an effort to frighten those unfamiliar with the subject, or to create a state of panic which might result in inadvisable legislation. The problem is really a serious one but as pointed out by A. S. Langsdorf, Director of Industrial Engineering and Research of Washington University, St. Louis, a great deal more information must be collected at numerous stations over wide areas at frequent intervals and over an extended period of time before any definite conclusions can be drawn. These data should be studied carefully in connection with health and other pertinent statistics and in co-operation with the medical profession.

Without intending to make light of so serious a subject, attention is called to the fact that without atmospheric pollution animal and vegetable life would not be possible upon this earth and some of the most awe-inspiring sights of nature would never occur. Without dust in the air to form the nuclei for rain drops, the entire earth would be constantly surrounded with enormous clouds of vapor, but it would never rain, and the beautiful sight of a rainbow would never exist. Without dust there would be no beautiful sunrise or sunset, and the northern climes would never know the wonders of the Aurora Borealis. Of course, the dust produced by nature would be sufficient to produce all these blessings and wonderful sights, and it would be desirable if domestic and industrial life could be so conducted that it would not add to the pollution of the air. These facts have been pointed out in order to coun-

teract to a small degree the publications of extremists who desire to hold up the question of atmospheric pollution as a kind of bugaboo.

The more progressive companies are keenly alive to the need for a reduction of air pollution. They are bending every effort of science, research, and experimentation toward the elimination of their part of the nuisance. If the records of past accomplishments are any indication of future progress, success will be achieved within a comparatively few years.

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DISCUSSION

H. E. BUMGARDNER (WRITTEN): In the special case cited by the author the smoke nuisance may have been abated but so far as cities generally are concerned the smoke abatement problem has only been scratched. Due to the present limitations of city smoke inspectors' work only a few scattered offenders are being apprehended and major offenders such as homes, laundries and other small industrial establishments are giving little or no attention to the smoke abatement problem.

In connection with the "means for measuring smoke density" mentioned in the paper I wish to call attention to the H.E.B. smoke chart shown in Fig. A which is different than any other chart heretofore devised.

In use, the chart is held in position with one hand for observing the smoke, and by moving the light shield or slide longitudinally of the slot with the other hand, the smoke may be observed through the slot in the chart. Thus the smoke being observed may be brought into direct juxtaposition with any one of the five shaded areas and when the observed smoke corresponds in color to the shaded area with which the sight opening is in registry, a reading may be taken directly from the chart. It will be noted that the shades of the various areas are marked adjacent to the slot so that after the observation is made an accurate reading may be readily obtained by noting the shade of the area on the chart adjacent which the sight opening is positioned.

XUM

By moving the slide with the sight opening across the slotted chart, the observer may match the shade of the smoke observed with one of the areas of the chart directly without the necessity of accommodating the eyes first to a chart and then to the smoke observed. The chart may be held substantially close to the eyes and an

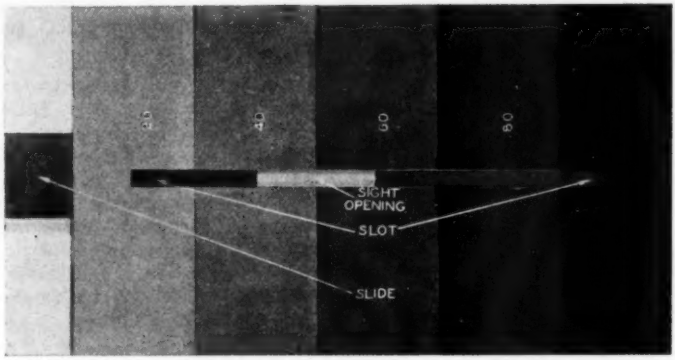

FIG. A. THE H.E.B. SMOKE CHART

accurate and fixed reading may be readily obtained. The principal advantages of this chart are:

(a) No white background surrounds the shaded areas as is the case with other charts
(b) It does not have to be placed at a distance from the observer or be revolved in order to produce shaded areas
(c) The smoke being observed and the shaded areas on the chart are brought into direct juxtaposition
(d) It provides a record of the smoke observed
(e) The shaded areas on the chart are conjoined
(f) It makes accurate smoke determinations easier and more convenient of attainment.

SABIN CROCKER (WRITTEN): This paper offers an interesting review of various means for removing objectionable materials from flue gases. Two angles of this problem touched upon by the author should be emphasized: (a) the human or educational side dealing with the awakening of sufficient public interest to make smoke elimination a live issue; (b) the prevention of smoke before its creation as distinguished from its elimination afterward.

In many cities there are laws against making excessive smoke, based on the Ringelmann scale, to which few outside the public utilities, large industrial concerns, and large office buildings or apartment houses pay any attention. Public interest is lacking, and the city smoke inspectors' office receives little backing and inadequate appropriations to carry on the work. In fact, the activity of most smoke inspection departments is confined to investigating complaints turned in by citizens who are particularly annoyed with smoke created by their immediate neighbors. Opinion is, in the main, apathetic toward the subject, and before much progress can be made in smoke abatement, the public must be sold on its advantages.

The principal offenders in creating smoke are the numerous users of soft coal on a small scale who employ hand firing and give little or no attention to smokeless combustion. Other prominent offenders are industrial plants engaged in the manu-

facture of powdered products such as lime, plaster, gypsum, cement and the like which allow excessive amounts of dust to escape into the atmosphere. The large power and heating plants which burn coal in underfeed stokers under expert supervision usually are distinguished by the absence of heavy smoke. An attempt to further reduce the smoke emission of such plants with the equipment described by Mr. Engle is hardly justified until progress has actually been made toward bringing the standards of the small users and industrial process plants above their present level. There is a field, however, for the equipment described by Mr. Engle in the catching of fly ash and cinders from pulverized coal firing where it already has been extensively applied.

Educational work dealing with smoke abatement should go further than merely arousing sentiment in favor of passing ordinances and enforcing their regulations. Constructive campaigns should be carried on to educate offenders in proper methods of hand firing, or sell them on the merits of smokeless fuel, the use of underfeed stokers and the like. Where bituminous coal is fired by hand, much can be done in an educational way to minimize the amount of smoke produced. The desirability of firing in thin layers and leaving part of the live fuel uncovered to burn the volatiles can be emphasized, and the advantage of admitting some preheated air over the fuel bed explained. Instruction of this kind in how to prevent smoke often will secure the cooperation of potential offenders where arrests and fines without such assistance merely would stir up antagonism to a smoke prevention campaign.

The use of oil, gas and anthracite fuels and combustion equipment which tend to minimize or eliminate smoke should be encouraged. Where bituminous coal must be burned with hand firing, an educational campaign should be carried on with a view to securing the best combustion possible. The down draft boiler assists materially in burning bituminous coal smokelessly. "Domestic" stokers also reduce smoke from low grade fuel. Some cities have drafted regulations governing the installation of such stokers with a view to obtaining enough volume in the combustion chamber to insure burning the volatiles before they are chilled by contact with water cooled surfaces.

The use of oil and gas fuel is to be desired, of course, from a smoke prevention standpoint, but often is precluded on account of their relatively high cost. These alternatives are pointed out as offering a satisfactory solution for small power and heating plants where the apparatus described by Mr. Engle would be entirely out of place. In fact, the use of such equipment is seldom required even in the largest plants to reduce smoke and dust emission within satisfactory limits, the principal exceptions being powdered coal firing and certain industrial process work.

As pointed out by the author, no suitable criterion is available by which equipment may be judged as to how thoroughly smoke is prevented or dust removed. A recent step in this direction, however, is provided by the development of a smoke recorder which embodies a photo-electric cell upon which a light is thrown. After penetrating the smoke-laden gas or dust filled air, the intensity of light striking the photo-electric cell permits the measurement of the quantity distribution of smoke or dust in the gas.

Any progressive step such as smoke abatement is of benefit to mankind primarily through reduction of smoke, and secondarily through stimulus to business and labor produced by the manufacture and sale of equipment required to prevent or abate smoke. If smoke abatement is worthwhile on its own merits, it may also be of assistance in a business way as one means of reducing unemployment. The line of thought prompting this suggestion was ably presented by Charles F. Kettering in the Dec. 13 issue of the *Saturday Evening Post*. While no mention of smoke prevention was made, the article did point out the need for more inventions such as the automobile, radio, domestic refrigeration to create new wants and hence new outlets for labor and capital.

XUM

H. B. MELLER (WRITTEN): Mr. Engle's paper is conservative, constructive and interesting; he is plainly unbiased in his treatment. The paper serves to emphasize the statement already made, that there is available adequate engineering knowledge to solve the problem of air pollution, in so far as emanations from stacks are contributory. The discussion of methods for prevention, separation or neutralization of the offending elements is most instructive and timely.

In considering visible smoke, Mr. Engle says: "It may be definitely stated, therefore, that the problem of smoke abatement can be solved by the selection of the proper fuel burning equipment and furnace design for the particular fuel to be burned, and by the proper operation of this equipment." This is true for the large plants which doubtless he has in mind; but there are the smaller plants for which there is no such satisfactory equipment—the hand-fired boilers and furnaces, the rank and file of heating plants, with their intermittent, inefficient firing. The need is for equipment which will put them in the same class, in point of smokelessness, with the larger, automatically fired and automatically regulated plant. That would seem to be one of the first necessary steps. The alternative is the use of a smokeless fuel.

The author properly divides dust into that which, under normal conditions, will settle out in a reasonable time, and that which will remain in suspension for longer periods. Of course the line of demarcation between the two classes varies with wind velocity, eddy currents, moisture content of the air, and other factors.

The dust collectors, separators and precipitators described in the paper are naturally, at this stage in the development of dust elimination, for the large plant. The need for proper equipment for the smaller plants can be expected to be met, in time, by the engineer. This time should be made as short as practicable.

In discussing noxious gases, Mr. Engle expresses the opinion that, where the sulphur content of the fuel is less than 2 per cent, it would be extremely difficult to trace any appreciable damage to human and vegetable life or to building materials to the discharge of such gases to the atmosphere at a reasonable height above the surrounding buildings. It seems to me that it is rather the total emission of sulphur gases that should be considered. If the sulphur content of the coal used were 2 per cent, for each million tons of fuel burned, with no provision for elimination there would be emitted something like 15,000 tons of sulphur, equivalent to 30,000 tons of sulphur dioxide (SO_2), or 37,500 tons of sulphur trioxide (SO_3), or nearly 46,000 tons of sulphuric acid (H_2SO_4). The fuel consumption in a large industrial city is measured in millions of tons per year. These products of the combustion of sulphur will come down, in one form or another, somewhere. Of course those from high industrial stacks will be spread over a much larger territory, but in number, most of the stacks are low, and the area of deposition would be less and the relative concentration greater. Meteorological conditions—rainfall, fogs, low wind velocities, etc.—also play a large part. A comprehensive study of the distribution of noxious gases was made in connection with the celebrated Anaconda smelter case, and is well worth reading by anyone interested.

I cannot agree that the problem of air pollution from stacks is not serious. It may be, as Mr. Engle says, that some writers—"extremists"—have held it up as a bugaboo. However, even if we were to ignore entirely the effects of the products of combustion upon vegetation and upon building materials, and to consider only the more obvious of the known effects upon the human system, there still would be, to my mind, ample reason for doing everything humanly possible to remedy the condition. There is no question that the outpouring from stacks aids in the formation and prolongation of fogs, thus accentuating any damage that may be done by smoke particles themselves; that it reduces the amount of sunshine, particularly in the beneficial shorter wavelengths; that it irritates the eyes, nose, throat and lungs, and may aggravate or cause diseases of those organs—these are a few of the effects that make it impossible to consider preventable air pollution as anything less than a menace.

As the author has indicated, and as I attempted to point out in my own paper, the first step in the abatement of smoke would seem to be better education in the selection and use of proper equipment and in the selection and use of fuel. A number of engineers have been and are studying the problem, and it is likely that the elimination of dust will first be studied in connection with the most efficient burning of powdered fuel. Mr. Engle's paper is a valuable contribution to the literature on smoke and dust abatement.

H. B. REYNOLDS (WRITTEN): The author presents a splendid review of the progress made in the development of apparatus for the reduction of air pollution.

There is no question that great strides have been made in recent years toward the reduction of the discharge of the larger particles classed as cinders, from stoker-fired plants. In fact, this problem has practically been solved by the development of several very efficient types of apparatus which have been described by Mr. Engle. However, the solution of the smoke and dust problem is far from a reality, even where the furnaces and fuel burning equipment have been designed along the most improved lines. The most modern power stations are too prone to smoke under the slightest provocation, such as a sudden increase of load, variation in the quality of coal, slight inattentiveness on the part of the operators, etc.

Mr. Engle spoke of the problem of the existing plants and called attention to some of the factors which can be changed or modified in an attempt to improve conditions. The owners and operators who have made progress in the solution of the smoke and dust problem in their plants should be encouraged to publish the results which they have accomplished and the methods they used in effecting the improvement.

As pointed out in the paper, very little is known as to the size of dust from stoker plants. However, if only that portion which falls within a few hundred feet of the plant is considered, then the size is considerably larger than that which will pass through a 200-mesh sieve, as in the case of one plant only about 1 or 2 per cent would pass through a 200-mesh sieve, while over 75 per cent would remain on a 40-mesh sieve.

I do not think it should be assumed that there is no problem in the case of fine fly ash simply because it travels a great distance before settling to the ground. It is true that the density of precipitation is much less than in the case of the larger particles, but the nuisance is extended over a large area. If the problem was one of density of precipitation, then there should be no objection to smoke, as smoke particles are so small that they no doubt travel for hundreds of miles, unless they are carried down by rain.

Mr. Engle points out a fact that is not generally appreciated, and that is that the large plants discharge much less carbon in proportion to the amount of fuel consumed, than do the small plants. If a large power station consuming 1,000 or more tons of coal per day, should turn out as much smoke in proportion to the amount of fuel consumed, as some small plants burning 2 or 3 tons a day, then the large plant would resemble a battle ship when laying down a smoke screen.

There is one important phase of smoke abatement work which should receive more attention from the engineers, and that is the framing of more reasonable and enforceable smoke laws, such as that which exist in Massachusetts and some other sections.

A. C. STERN (WRITTEN): Historical reviews of this nature are valuable every few years to sum up progress and eliminate repetition of laborious development. The author was wise in not giving efficiency curves of the various methods discussed, for the reason that he himself mentions; the lack of a fair standard of comparison. I, however, disagree with his suggested remedy of this lack by using standardized fabric filters. According to my viewpoint the solution is to develop methods of sampling the gas stream before and after the units under test, that will

allow size distribution curves to be drawn for the sub-sieve in addition to the sieve sizes.

The author quotes Gibbs' classification for smoke and dust but he includes in the latter the class termed by Gibbs "cloud." THE AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS is, however, more familiar with the classification of Drinker into dust $150\mu - 1.0\mu$ fume: $0.2\mu - 1.0\mu$ and smoke $<0.3\mu$. At the risk of confusing the classification still more, I should like to suggest the use of the terms smoke, fume and dust only as names descriptive of sources and methods of formation rather than size limits. If size limits are desired, couple them with such terms as appear frequently in Mr. Engle's and other papers, as fine, coarse and intermediate dusts, etc. These limits may be far into the sub-sieve range since it is certainly high time that we obtained more specific measures than "95 per cent through 200-mesh sieve." Methods of measurement are available that will give these results and need only receive more publicity before they will be adopted in the measurement of pulverized coal and fly ash.

I take exception to Mr. Engle's proof that the finer sized particles are relatively not objectionable because of the large area over which they are spread, because I believe that the relative number of such small particles is so many times greater than the number of coarse particles as to cause a potential menace. This is particularly true of the $\frac{1}{10}\mu$ to 10μ range that is retainable in the lungs and may be harmful if composed of broken ash bubbles.

In his description of electrical precipitation, the statement is made that, "the high potential ionizes the dust-laden gas etc." I believe that it is more technically correct to say that the dust particles, in order to be precipitated, must have been previously ionized before entering the precipitator. In the case of fly ash, the ionization is thermal.

The effect of rainbows and aurora are caused and have been caused for centuries by matter that can not in the strict sense of the word be at all considered as atmospheric pollution.

COL. ELLIOTT H. WHITLOCK (WRITTEN): Mr. Engle has used in his paper some figures about the contamination of the atmosphere in New York City, giving Dr. Free as his authority. So much is not known at present about methods and schemes for sampling and the various schemes used gives such variable results that unless we know what method Dr. Free used, the figures quoted have but little or no meaning.

These figures do not check with the figures recorded when using an Owens' automatic dust counter for weight per volume, nor does the percentage of soot content check within any reasonable figure of error with results obtained when using the so-called "soot-fall" test.

It would seem to me that some further effort should be made to establish a reasonable and practical method for the collection of uniform samples and of the proper interpretation of these samples so that results given to the public may carry with them a reasonable degree of accuracy.

PRESIDENT HARDING: We are indebted to Mr. Engle for making such a fine resumé of the history of the art of dust separation. It is a fine reference paper for all engineers interested in problems of this character.

I do know that the scrubber type washer, which any one can construct, will remove a sufficient amount of the dust from agricultural limestone plants and cement plants so that it will pass the usual ordinance and will stop complaints from the neighbors who live adjacent to these plants. The apparatus is not too expensive to install, but, of course, it costs money to operate the necessary fan and pumps. If you install enough eliminator plates, provided with wash down sprays and make them out of cypress, you can collect the objectionable dust.

No. 893

SPECTRAL DISTRIBUTION OF THE ENERGY RADIATED FROM METALLIC SURFACES AT HIGH TEMPERATURES

By G. R. GREENSLADE,[1] PITTSBURGH, PA.

NON-MEMBER

This paper contains information of scientific value on the subject of radiation from metallic surfaces. While the data are not directly applicable to the problems of the heating and ventilating engineer, they are of considerable value in connection with certain phases of heat emission by direct radiation. It is contemplated that this paper will form the basis for additional work at the A.S.H.V.E. Laboratory on other metallic surfaces in which the heating engineer is interested.

THE work of determining the exact nature of the thermal radiation from the polished surfaces of metals was begun by Paschen[2] who in 1896 published the results of an investigation which seemed to indicate that the spectral distribution of the heat radiation from certain metallic surfaces at temperature T degrees Kelvin (centigrade absolute scale) could be expressed by the following equation:

$$E_{\lambda T} = C_1 \times \lambda^{-a} \times e^{-C_2/\lambda T} \qquad (1)$$

where

λ = the wavelength of the radiant energy E

C_1, C_2, a = constants depending upon the nature of the metal under consideration.

In 1899, Lummer and Pringsheim[3] experimentally obtained a number of spectral energy distribution curves for platinum covering a fairly wide range of temperatures. On attempting to check Paschen's equation by means of these

[1] Chief Physicist, Flannery Bolt Company, Pittsburgh, Pa.
[2] Wiedemann; *Annalen der Physik und Chemie*, 58-455-1896; 60-662-1897.
[3] *Verhandlungen der Deutschen Physikalischen Gesellschaft*, 1-215-1899.
Presented at the 37th Annual Meeting of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, Pittsburgh, Pa., January, 1931.

data using $a = 6$ they found that the theoretical curves dropped off more rapidly at both extremes than did the experimental ones, thus indicating that the equation was inadequate for use over more than a very narrow range of wavelengths. Subsequent to this Aschkinass,[4] by means of Planck's[5] black body radiation law, Kirchoff's law, and the relation which Maxwell's electromagnetic theory indicates should exist between electrical resistance and reflecting power, derived the following equation which it was thought might be applied to all metals:

$$E_{\lambda T} = C_1 \times 0.0221 \sqrt{\rho_0 T}\, \lambda^{-5.5} \times \frac{1}{e^{\frac{C_2}{\lambda T}} - 1} \qquad (2)$$

The symbols used are the same as those previously used with the exception of the new quantity ρ_0, which represents the specific resistance of the specimen at zero degrees centigrade.

This equation agreed with the experimental curves for platinum obtained by Lummer and Pringsheim much better than did equation (1), and accordingly it was incorporated in an article by O. Lummer[6] and republished, together with the curves, showing the experimental agreement. There are, however, two points in the derivation which make the extensive applicability of the equation questionable. The first of these is that it is based on the Maxwell relation mentioned, i. e.

$$100 - R_\lambda = \frac{36.5}{\sqrt{\lambda}} \times \sqrt{\rho} \qquad (3)$$

where

R_λ = reflecting power of the metal for radiation of wavelength λ

ρ = its specific resistance.

This theoretical equation was found by Hagen and Rubens[7] to hold only for very long wavelengths of the order of 12μ or greater, while for wavelengths of 4μ their work showed that in the determination of the values of the quantity.

$$C_\lambda = \frac{100 - R_\lambda}{\sqrt{\rho}} \qquad (4)$$

for the various metals and alloys used there was an average deviation from the mean of 21 per cent. The second questionable step referred to is that the introduction of ρ_0 in the equation, involved the assumption of complete linearity for the temperature function of resistance.

A number of years after the publication of the investigations of Lummer and Pringsheim, G. V. McCauley[8] repeated essentially the same work that

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[4] *Annalen der Physik*, 17-960-1905.
[5] *Vorlesungen uber die Theorie der Warmestrahlung*, Leipzig, 1906.
[6] *Handbuck der Elektrizitat und des Magnestisms*, Band 2, Lief, 2-462-1914.
[7] *Annalen der Physik*, 11-873-1903.
[8] *Astrophysical Journal*, 37-164-1913.

they had done on platinum using apparatus so constructed that the beam of radiant energy passed through practically no absorbing medium from the time of its emission by the metallic surface up to the time of its absorption by the bolometer which measured its intensity. This was a distinct improvement over any method which had been used before and gave results which were more nearly free from absorption bands. McCauley also obtained comprehensive sets of curves for both palladium and tungsten.

The purpose of carrying on the experimentation described in this paper was to extend the data in this field so as to include analyses of the radiation spectra of a number of metals not previously investigated, in order that further tests might be made of the applicability of some of the theoretical distribution equations which have been evolved. The form of apparatus and the methods employed in the present work were somewhat similar to those used by McCauley, though they have been modified in many details so as to eliminate certain difficulties which he encountered. A number of curves were obtained for each of the following metals: Tungsten, iron, molybdenum, nickel, cobalt, iridium, rhodium and gold, and in addition to this, the work on platinum was repeated for comparison purposes.

APPARATUS

The general arrangement of the apparatus as used in this analysis is shown in Fig. 1. The specimen to be tested was placed at S_1 in the water cooled vacuum radiation chamber V_1 and was heated electrically by leads sealed into the base of the chamber. The radiation was allowed to pass through the rock salt window W_4 into the large vacuum spectrometer chamber V_2 by opening the shutter S_2. The path of the rays through this analyzing chamber has been indicated by light lines in the figure. A concave mirror M_2 was used to focus the beam on the slit S. After having passed this slit it fell on the collimating mirror M_3 and then suffered dispersion by the rock salt prism shown at the center of the figure. It was next reflected from the Wadsworth [9] mirror M_4 to the concave mirror M_5 which in turn brought it to a focus at the exposed strip of the bolometer B. Since both of the chambers were evacuated, and all mirrors of the spectrometer were silvered on their exposed surfaces the beam of radiant energy passed through no material medium except rock salt, which, for the wavelengths considered has almost perfect transmission. By this means the loss of energy by absorption was reduced to a minimum and in all cases where it was not negligible, corrections were computed and applied to the data.

Four pumps were used for exhausting the two vacuum chambers. Three of these, indicated diagrammatically in Fig. 1, were of the following types: P_1 a Gaede vane pump, P_2 a Gaede mercury pump, and P_3 a rotary molecular pump, built by J. P. Foerst, Mechanician, University of Wisconsin. The fourth pump, not shown in the diagram, was of the rotary oil immersed variety and was placed in the vacuum line of the spectrometer chamber, for the purpose of securing better control of the pressure in the space around the bolometer.

The preparation of the test specimens was as follows: A sheet of the metal, whose spectral energy distribution was desired, was rolled out until it was about 0.04 mm thick. After this it was carefully polished and cut to size, usually about 12 mm wide and 50 mm long. It was then folded over into a black body

[9] *Philosophical Magazine*, 38-337-1894.

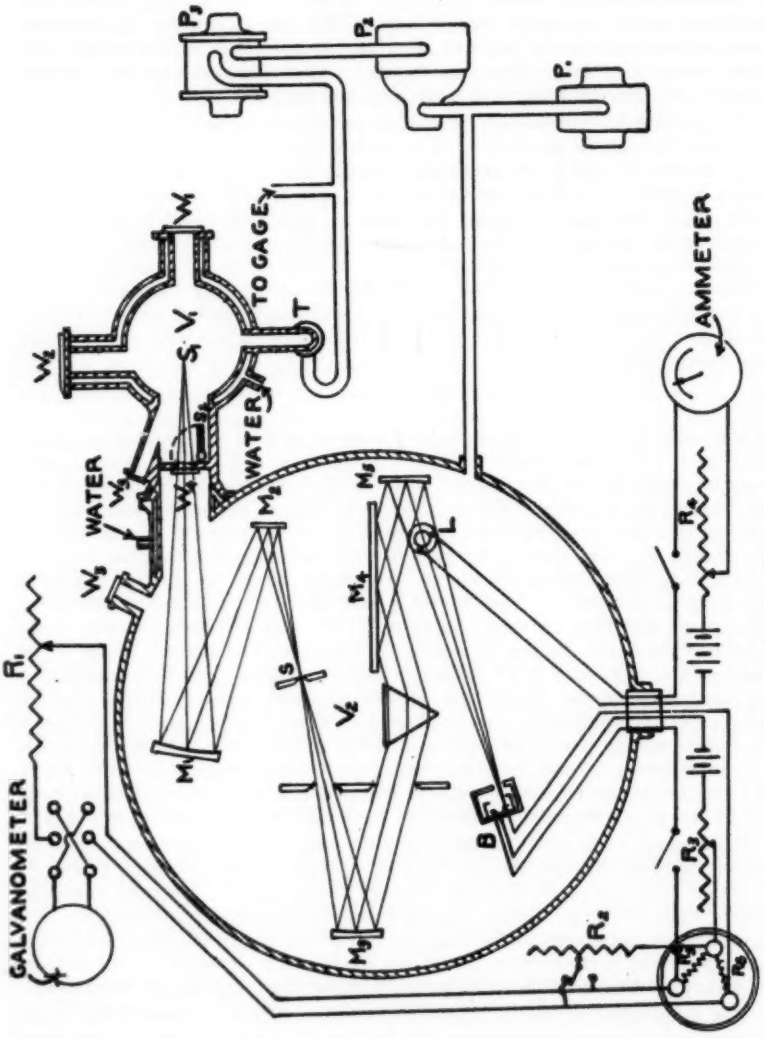

FIG. 1. ARRANGEMENT OF APPARATUS

wedge [10] having an angle of 12 deg. and was mounted vertically by clamping its ends in special machined blocks so constructed as to keep the angle of the V fixed. These clamps served also as current leads for electrically heating the radiating strip to any desired temperature. The lower one was equipped with a mechanism for moving it slowly up or down in order to compensate for temperature variations in the length of the strip.

The orientation of the specimen when in position was such that the opening of the black body V was in the direction of the glass window W_2, while one of the flat polished surfaces faced the aperture leading into the spectrometer chamber. In this position the true temperature of the metallic strip could be determined with an optical pyrometer, by sighting into the black body crevice at the same time that the bolometric apparatus was being used for measuring the characteristic emission which the polished surface was radiating to the analyzing apparatus.

The length of specimen necessary to form an image of sufficient magnitude to cover the entire opening of the spectrometer slit was a little more than a centimeter, but as the clamped ends were cooled to some extent by conduction, it was essential to use a considerably greater length than this in order that the entire portion which illuminated the slit might be at the same temperature. After deducting the amount covered by the clamps, the specimens which were used still presented a radiating surface of about 4.5 cm length, or approximately four times the length required to fill the slit. This size usually gave very satisfactory results, but in some cases, a difference of several degrees could be detected by means of the optical pyrometer when exploring a section of one centimeter length chosen near the center of the specimen.

The width of the spectrometer slit was 0.312 mm, this being the opening whose monochromatic image would just cover the exposed strip of the bolometer. The spectra resulting from an aperture of this magnitude are, of course, somewhat impure and in consequence give wave-length-energy curves which are a little too flat in regions of great curvature; consequently it is necessary in instances where the final results are affected appreciably, to reduce the curves to a pure spectrum basis. The manner of making this reduction in the present work was by means of the graphical method which has been worked out by Runge.[11]

The bolometer was mounted in a double walled brass shell, which had its annular space filled with water through an opening which was afterwards sealed. This served the purpose of giving it a large heat capacity, thus eliminating any rapid changes of temperature, which might affect the dependability of the apparatus. In order to secure uniformity in regard to radiation and convection constants, the exposed and shielded strips of the bolometer were made as nearly alike as possible, each being composed of blackened platinum foil 0.002 mm thick, 0.48 mm wide, and 12 mm long, and each having a resistance of approximately 2.6 ohms. These strips formed two of the arms of a Wheatstone Bridge and were connected to the rest of the circuit as shown in Fig. 1. In order to minimize the effect of temperature changes on the resistance of the two external arms of the bridge R_5 and R_6, they were constructed of manganin wire, wound on a water-filled metal cylinder 10 cm in diameter

[10] C. E. Mendenhall, *Astrophysical Journal*, 33-91-1911.
[11] *Zeitschrift für Mathematik und Physik*, 42-205-1897.

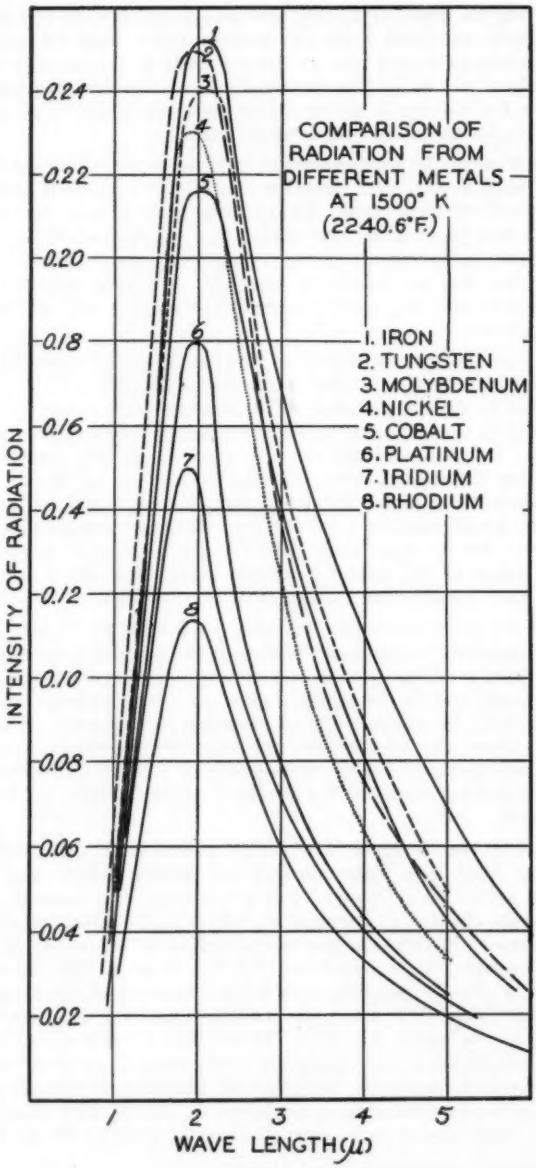
COMPARISON OF RADIATION FROM DIFFERENT METALS AT 1500° K (2240.6°F.)
1. IRON
2. TUNGSTEN
3. MOLYBDENUM
4. NICKEL
5. COBALT
6. PLATINUM
7. IRIDIUM
8. RHODIUM
INTENSITY OF RADIATION
0.24
0.22
0.20
0.18
0.16
0.14
0.12
0.10
0.08
0.06
0.04
0.02
1
2
3
4
5
WAVE LENGTH (μ)

FIG. 2.

and 10 cm long, and were immersed in a tank of oil. Each of these coils had a fixed resistance of approximately 17.7 ohms. The balancing of the bridge was accomplished by means of a large variable manganin resistance R_2 so arranged that it might be thrown in parallel with either R_5 or R_6. A high sensivity shielded galvanometer connected across the bridge was used for determining this balance and also for measuring, by the magnitude of the deflection from the position of balance, the intensity of the radiation of the wavelength interval which was caused to fall on the exposed bolometer strip by the opening of the shutter S_2 in the radiation chamber. A complete description of this galvanometer is given in an article by Weniger [12] on Absorption Spectra and it will, therefore, be sufficient to state here that it was of the moving magnet type, had a current sensivity of 5×10^{-10} amperes per mm at a distance of one meter, and was adjusted so as to be dead beat. Whenever possible the instrument was used with full sensitivity, but when working at high temperatures, or at wavelengths in the vicinity of maximum emission, it was necessary to reduce this by switching in one of a number of definite series resistances. The following is a list of the factors by which it was found necessary to increase the galvanometer readings in order to reduce to the same basis the data taken on the different sensitivity steps:

First sensitivity— 1.00 × deflection
Second sensitivity— 2.68 × deflection
Third sensitivity— 4.83 × deflection
Fourth sensitivity— 9.15 × deflection
Fifth sensitivity—17.93 × deflection
Sixth sensitivity—36.22 × deflection

These factors were determined by taking a large number of comparison readings over the entire period during which this experimental work was carried on, several of them being taken each time that a run on a specimen was made, in order that no change might creep in without being detected.

THE STANDARD LAMP

The entire bolometer and galvanometer system was standardized by means of an electric comparison lamp, shown at L in Fig. 1. The purpose of placing it at this position inside of the spectrometer chamber was to have it instantly available for checking the energy measuring system at any time during a run, without the necessity of disturbing any of the apparatus, or even cooling the heated test specimen. The lamp was of the small low voltage carbon variety having a single loop filament, such as is commonly used for optical pyrometric work. After having been carefully aged this lamp was mounted inside of a series of coaxial cylindrical shields in such a manner that its radiation was completely cut off from the apparatus in the chamber with the exception of a small beam from the central portion of the filament which was allowed to shine on the bolometer strip by passing through a series of apertures in the shields previously mentioned. The radiation from the standard lamp was cut off by means of an electro-magnetic shutter which closed the outer aperture. Any desired variation in the intensity of the light was produced by means of the control resistance R_4.

[12] *Physical Review*, 31-393-1910.

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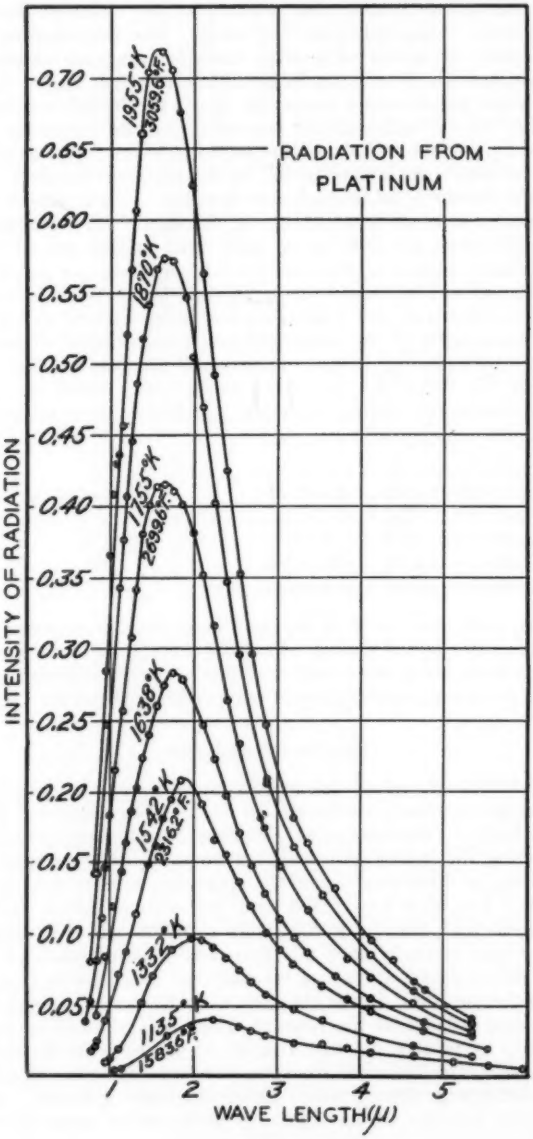
RADIATION FROM PLATINUM
INTENSITY OF RADIATION
WAVE LENGTH(μ)
0.70
0.65
0.60
0.55
0.50
0.45
0.40
0.35
0.30
0.25
0.20
0.15
0.10
0.05
1
2
3
4
5
1953° K
3059° F.
1870° K
1755° K
2696° F.
1638° K
1542° K
2316° F.
1332° K
1135° K
1583° F.

FIG. 3.

In order to obtain a relation between filament current in the standard lamp and intensity of radiation falling on the bolometer, a large tungsten ribbon lamp was mounted just outside of the window W_1 with a lens so placed as to produce an image of the strip at the position S_1, which is normally occupied by the black body V. The ribbon lamp was heated to an arbitrarily chosen brillancy, and its effect on the bolometric system obtained by opening the shutter S_2. As soon as the galvanometer deflection had been read, S_2 was closed and a deflection of the same magnitude was again produced by direct radiation from the lamp at L. After the filament current necessary for this had been recorded, a rotating sector was placed between the ribbon lamp and the window W_2 in order to reduce the energy to a definite fraction of its original value. This comparison was again made and by continuation of the process, using a number of sectors, sufficient data were obtained to plot a curve representing filament current as a function of radiation intensity. Upon examination this graph proved to be a parabola and consequently the following relation holds:

$$R = K\,(I - I_0)^2 \tag{5}$$

where

R = intensity of radiation,
K = a constant,
I = current at any time,
I_0 = value of I at the point of tangency of the curve and the current axis.

The point I_0 was obtained accurately by plotting the square root of the radiation intensity, against filament current and taking the intercept on the current axis. This gave $I_0 = 0.193$ amp. All radiation curves for the different metals were reduced to the same energy scale, but since this scale was arbitrarily chosen the constant K of the calibration equation was dropped out, thus reducing the relation to

$$R = (I - 0.193)^2 \tag{6}$$

PROTECTION OF THE METAL SPECIMENS

As the presence of active gases and vapors is damaging to most metallic filaments at high temperatures, special methods for gas elimination were used throughout this work. By means of the series of vacuum pumps mentioned above, it was possible in most cases to keep the pressure in the radiation chamber down to 0.0001 or 0.0002 mm of mercury. At the highest working temperatures, however, the vacuum fell somewhat below this value on account of the *out-gassing* of the inside walls of the chamber due to superficial heating by absorbed radiation.

In order to eliminate grease and oil vapors as far as possible, the opening in the bottom of the radiation chamber for the replacement of specimens was rendered gas-tight by means of lead gaskets instead of wax. The liquid air trap indicated at T in Fig. 1 served to prevent the diffusion of oil and mercury vapors from the pumps and pressure gage back into the radiation chamber. This trap was made of metal and was connected by means of metal tubing directly to the radiation chamber in order that any vapors from the wax joint between it and the glass vacuum line might be kept away from the heated

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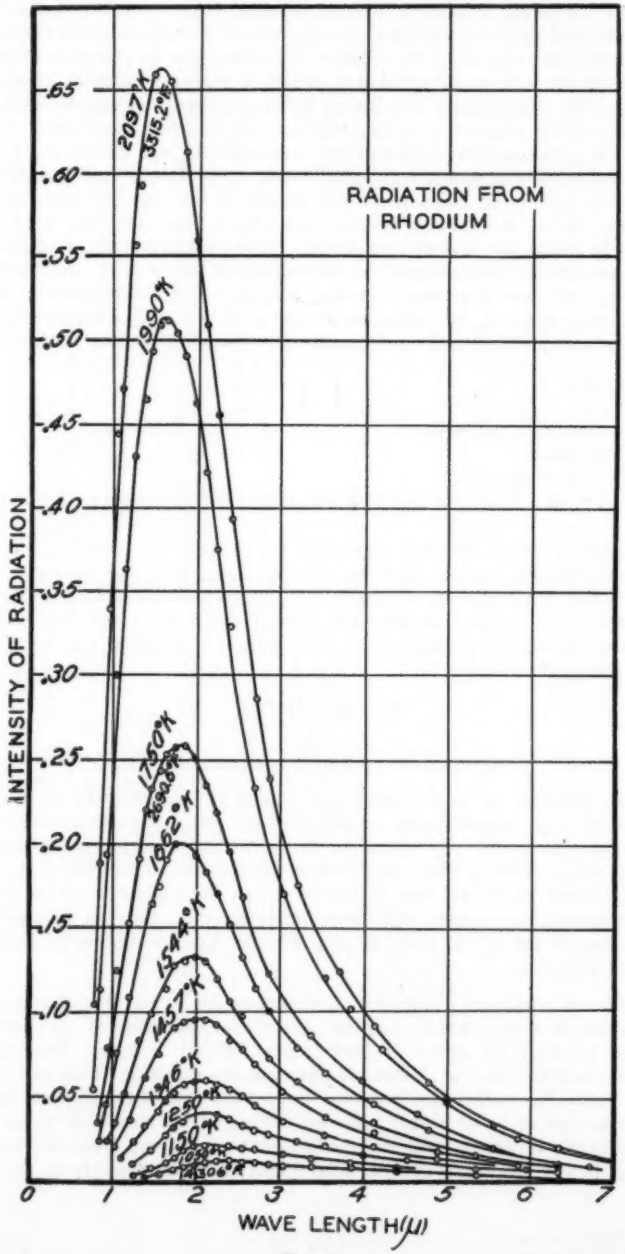
RADIATION FROM
RHODIUM
INTENSITY OF RADIATION
WAVE LENGTH(μ)
2097°K
1990°K
1750°K
1662°K
1544°K
1457°K
1346°K
1250°K
1150°K
1030°K
.65
.60
.55
.50
.45
.40
.35
.30
.25
.20
.15
.10
.05
0
1
2
3
4
5
6
7

FIG. 4.

specimen. During all runs this trap was kept immersed in a Dewar flask of liquid air.

The removal of water vapor from the radiation chamber was accomplished in the ordinary manner by means of phosphoric anhydride. It was found that in the use of this compound, one precaution was necessary in order to avoid shortening the life of the test specimen; namely, that of shielding the surface of the drier from intense direct radiation. If this precaution were not observed the inside wall of the chamber as well as the glass and rock salt windows would become coated with syrupy phosphoric acid, and the specimen, if of an oxidizable material, would disappear rapidly. The probable action that takes places in such a case is that the globules of orthophosphoric acid present on the surface of the anhydride are reduced by the strong radiation to metaphosphoric acid, which in turn evaporates and after being condensed on the cool inside walls of the chamber again takes up the water previously driven off, thus forming once more the viscous semi-solid compound. The destruction of the filament is evidently accomplished by some cyclical oxidizing and reducing process resulting from the dissociation of the vapors which come in contact with the hot metal. This phenomenon only occurred during runs at very high temperatures and was completely eliminated in all cases by the use of a thin metal shield to cut off the direct radiation from the drier.

General Precautions

The purpose of evacuating the large chamber V_2, Fig. 1, containing the spectral apparatus, was principally for the elimination of the absorption of energy by water vapor, carbon dioxide, and oxygen of the air. Consequently it was not necessary in this case to maintain as high a vacuum as was required in the radiation chamber. It was, however, found to be very essential, when using a low working pressure, to keep it absolutely constant since, as is well known, the sensitivity of the bolometer for pressures less than one millimeter of mercury is very materially affected by small variations of this nature. Throughout the present work the pressure in this chamber was kept constant at 0.1 millimeter of mercury.

Since water vapor is removed very slowly by the ordinary vacuum pump, a further assurance of its elimination was obtained by placing open trays of phosphoric anhydride and also of concentrated sulphuric acid at various places inside the vacuum chamber.

In order to guard against the possibility of absorption, due to the collection of a film of water on the surface of the rock salt window W_4 when preparing it under conditions of high atmospheric humidity, a convenient method suggested by M. H. Petersen was followed; namely, that of grinding and polishing under toluene. During all of this process the further precaution was observed of using rubber gloves when handling the window, in order to prevent any clouding of the polished surface by moisture from the fingers. When the polishing was completed, the window was put in place as quickly as possible and was kept protected on both sides by the use of phosphoric anhydride. Whenever the radiation chamber was opened for the purpose of replacing specimens, it was filled with dry air, and then, during the time of mounting up the new metal strip, the hole in the base of the chamber was kept closed by means of a temporary cover for the elimination of air currents.

Contamination of the silvered surfaces of the spectrometer mirrors by the diffusion of mercury vapor into the chamber from the vacuum line was prevented by a tube 6 cm in diameter and 20 cm long, which was packed loosely with a mixture of glass wool and gold leaf and was attached to the inside wall of the chamber at the point where the glass vacuum line entered. In this way any vapors coming into the spectrometer chamber were caused to flow over a large area of gold surface before reaching the apparatus inside.

TEMPERATURE MEASUREMENTS

As has already been stated, the true temperatures of the metals studied were determined by means of an optical pyrometer. The instrument used for the purpose was built by J. P. Foerst, mechanician, and was very similar to one described in the Astrophysical Journal.[13] It was a modification of the Morse type employing rotating sectors to adapt it for use over wide ranges of temperature, and selective transmission screens for obtaining the color match necessary when computing the results by means of Wien's distribution equation. For all except the lowest temperatures measured, a green filter consisting of three layers of gelatin screen was used. The visible transmission of this screen, for black body radiation of a certain temperature, was obtained by first analyzing it spectrophotometrically[14] and then, after making the necessary reduction to black body distribution, combining the results so obtained with Ives[15] curve for the sensitivity of the eye as a function of wave-length. By using different temperatures, a number of these transmission graphs were plotted, from which, by means of a planimeter, the following effective wavelengths of transmission were obtained:

λ (Effective) = 0.5362 μ for 1000 K.
λ (Effective) = 0.5355 μ for 2000 K.
λ (Effective) = 0.5351 μ for 3000 K.

It may be seen from these results that temperature changes produce only slight shifts in the visible transmission of the screen. This characteristic is due largely to the narrowness of the transmission band of the material. It transpires also that these results are nearly independent of lack of precision in the value taken for the quantity C_2 of Wien's equation as used in making the reduction to black body distribution mentioned. By assuming first that this constant is equal to 14300, and then to 14570, a difference of less than $5 \times 10^{-5} \mu$ is produced in the computed values of the effective wavelengths of transmission. W. E. Forsythe reported several years ago[16] on some green pyrometer screens made of glass, which seemed to be somewhat unsatisfactory due apparently to the broadness of the transmission band, for he found that when using reduction sectors of small aperture the nature of the light coming from the source was so different from that emitted by the comparison lamp that a good color match could not be obtained. Very little difficulty of this nature was experienced in

[13] W. E. Forsythe, *Astrophysical Journal*, 43-295-1916.
[14] Two specimens of the screen, procured from the manufacturers at different times, were analyzed independently for transmission by Dr. C. E. Mendenhall and by the author, and were found to agree in effective wave-length of transmission to 0.0002μ.
[15] *Philosophical Magazine*, 24-860-1912.
[16] *Journal of the Optical Society of America*, 5-85-1921.

the case of the gelatin screens used in the present work and it is believed that their transmission is sufficiently narrow to make them somewhat superior in this respect to the green glass examined by Dr. Forsythe. It is possible, however, that in regard to permanence, the gelatin screens may not be as good as those made of glass containing a body color.

For the measurement of temperatures below 1500 K it was found to be in-

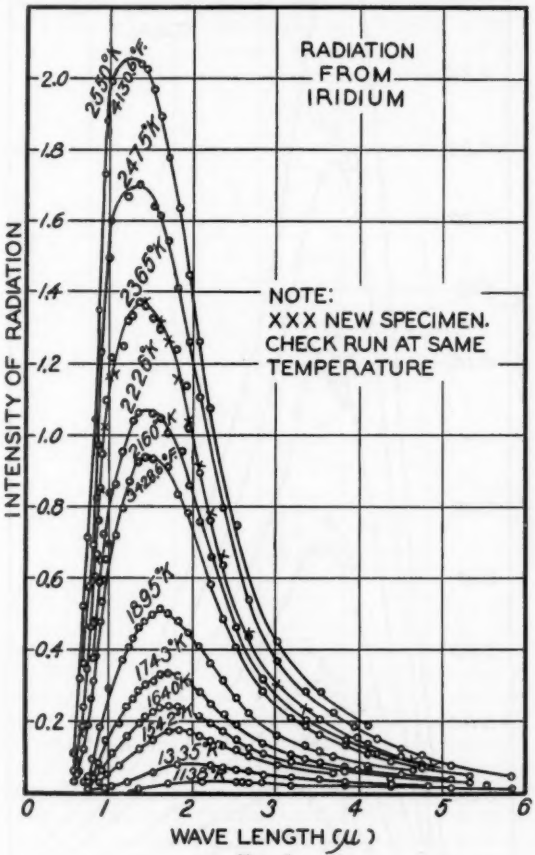

FIG. 5.

feasible to make use of green filters on account of the small amount of green light emitted by the source. In such cases a standard red screen of Jena glass No. F-2745 was employed. The effective wavelength of transmission of this filter for the range of temperature used was obtained from the work of Hyde, Cady, and Forsythe [17] of the Nela Research Laboratory.

[17] *Astrophysical Journal*, 42-299-1915

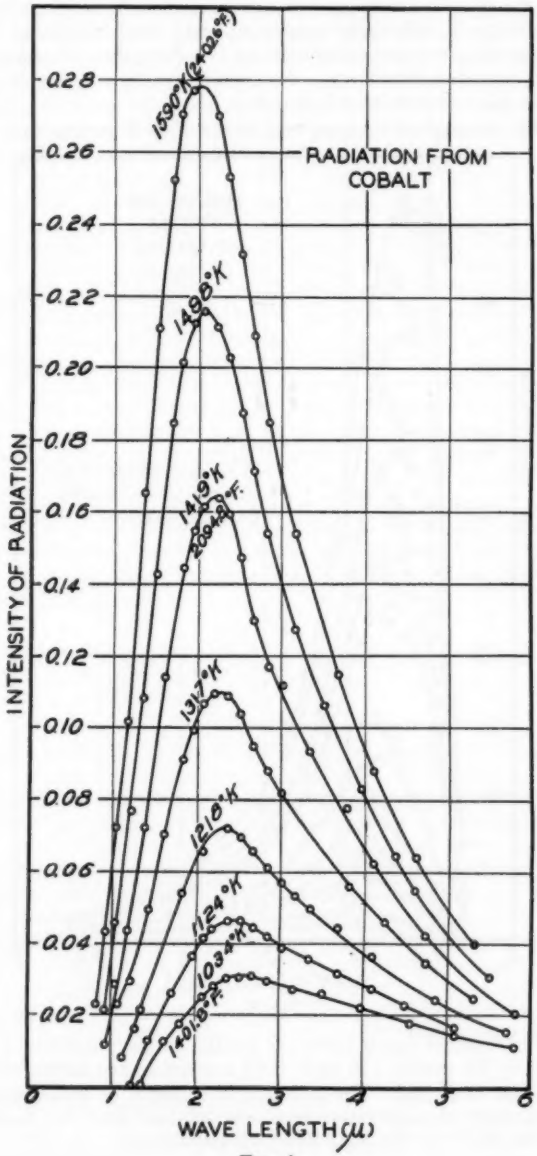
RADIATION FROM COBALT
INTENSITY OF RADIATION
WAVE LENGTH (μ)
0.28
0.26
0.24
0.22
0.20
0.18
0.16
0.14
0.12
0.10
0.08
0.06
0.04
0.02
0
1
2
3
4
5
6
1590°K (2402°F)
1498°K
1419°K
2004°F.
1317°K
1210°K
1124°K
1034°K
1401.5°F.

FIG. 6.

The optical pyrometer was calibrated by using rotating sectors for changing the intensity of a beam of light coming from a black body furnace, of a type described elsewhere,[18] which was maintained at constant temperature. The value selected for this in the case of the green calibration was the melting point of palladium which was assumed to be 1822 K as determined by Day and Sosman.[19] In the case of the red calibration for low temperatures, the gold point (1336 K) was used. These standardizing temperatures were obtained by running a number of melts inside the black body chamber of the furnace, making use of what has been termed the *wire method*. In spite of the fact that the

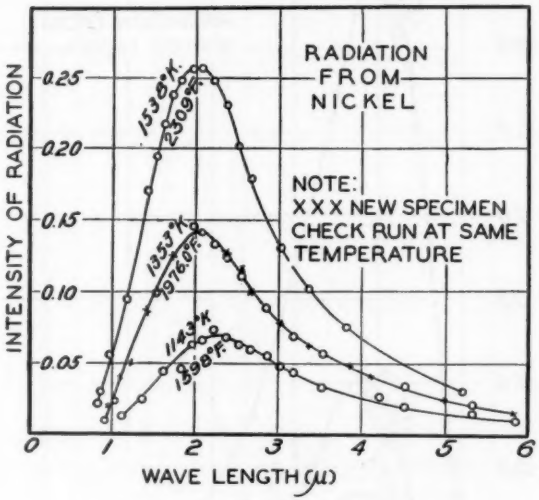

FIG. 7.

pyrometer comparison lamp was carefully *aged* at full brilliancy for a period of several days before being calibrated, it was found to change gradually with use, thus necessitating an occasional recalibration. This was made every two weeks during the time that the instrument was being employed. On account of this lamp variation some of the temperature measurements are probably somewhat in error. This error is negligible for the lower determinations, perhaps one degree at 1400 K, and possibly as much as 10 deg. at 3000 K. The comparison lamp used in the pyrometer was especially constructed for the purpose, being made with a single 4-mil tungsten filament having a slight kink near the middle for use in locating the point at which the color match was to be secured. When the pyrometer was used with the green filter the current required by the comparison lamp in covering the necessary temperature range varied from 0.63 to 0.97 amp., while with the red glass screen the variation was from 0.42 to 0.55 amp. Thus the largest current used was well below 1.5 amp., the rated full current for the lamp.

[18] C. E. Mendenhall, *Physical Review*, 10-518, 519-1917.
[19] *American Journal of Science*, 29-93, 161-1910.

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SUMMARY AND CONCLUSION

The radiation curves for the different metals (Figs. 2 to 12, inclusive) have been corrected for galvanometer drift, impurity of spectrum due to width of

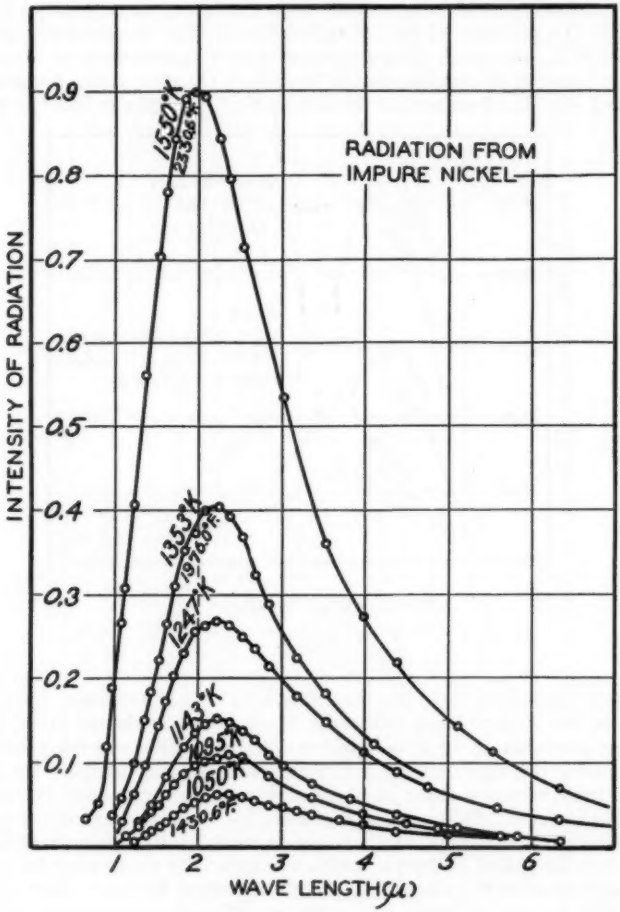

FIG. 8.

spectrometer slit, change of bolometer sensitivity due to slight pressure variations in the vacuum chamber, absorption of energy by the rock salt window and prism, and for variations in spectral concentration due to the non-linear dispersion of rock salt. Corrections, to the data, have also been applied for change in the reflecting power of the silver mirrors as a function of radiation

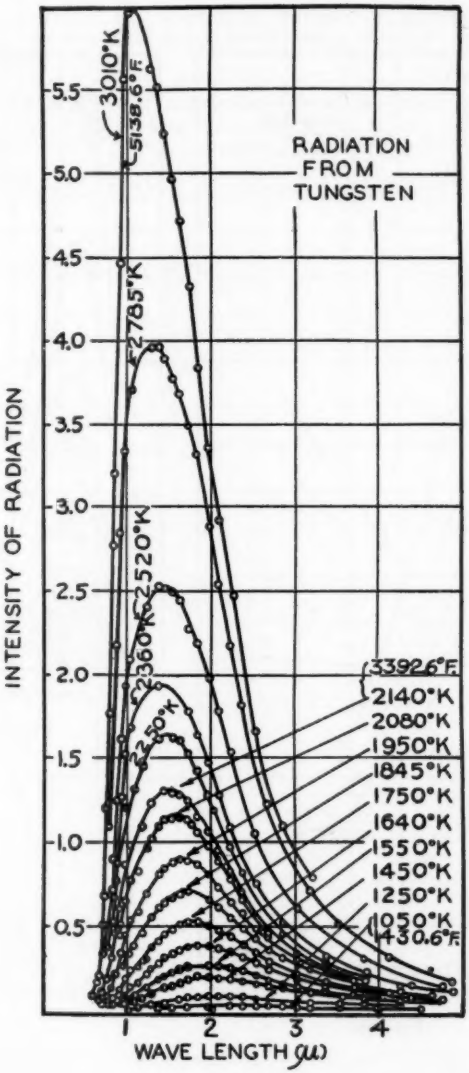
RADIATION
FROM
TUNGSTEN
INTENSITY OF RADIATION
WAVE LENGTH μ
3010°K
5138.6°F.
2785°K
2520°K
2360°K
2250°K
3392.6°F.
2140°K
2080°K
1950°K
1845°K
1750°K
1640°K
1550°K
1450°K
1250°K
1050°K
1430.6°F.
5.5
5.0
4.5
4.0
3.5
3.0
2.5
2.0
1.5
1.0
0.5
1
2
3
4

FIG. 9.

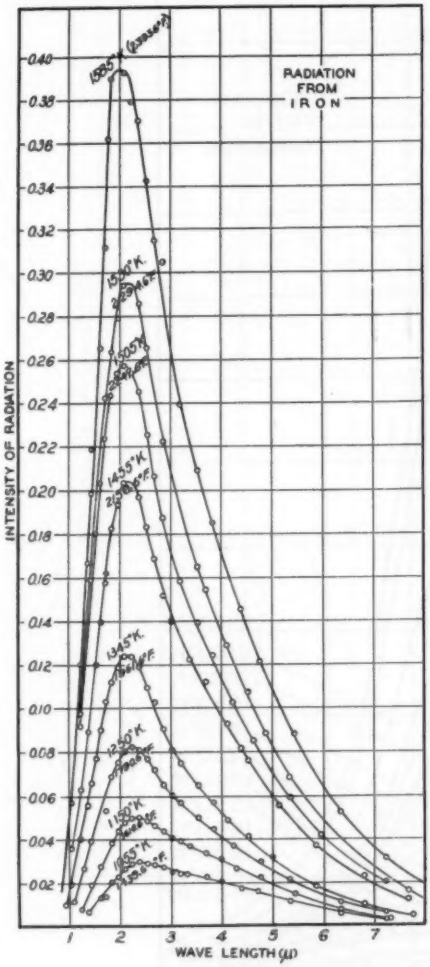
RADIATION FROM IRON
INTENSITY OF RADIATION
WAVE LENGTH μ
0.40
0.38
0.36
0.34
0.32
0.30
0.28
0.26
0.24
0.22
0.20
0.18
0.16
0.14
0.12
0.10
0.08
0.06
0.04
0.02
1 2 3 4 5 6 7

FIG. 10.

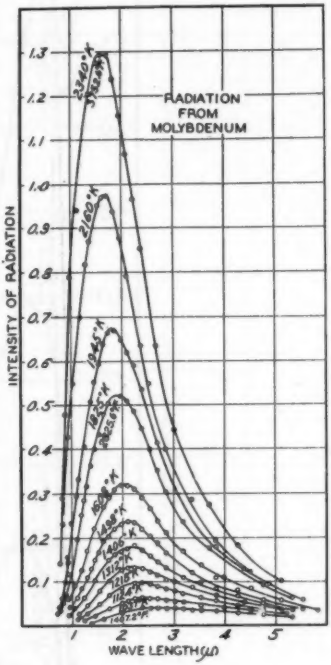
RADIATION FROM MOLYBDENUM
INTENSITY OF RADIATION
WAVE LENGTH μ
1.3
1.2
1.1
1.0
0.9
0.8
0.7
0.6
0.5
0.4
0.3
0.2
0.1
1 2 3 4 5

FIG. 11.

wavelength, and for change in the reflecting power of the surfaces of the rock salt prism as a function of the angle of incidence of the beam. Furthermore, the curves have been reduced to a common energy basis by comparison with an arbitrary standard source.

It was found that the total radiated energy, as well as the form of the wavelength-energy curve varied with the purity of the metal. The extent of this variation may be seen from the following: one specimen of commercial nickel

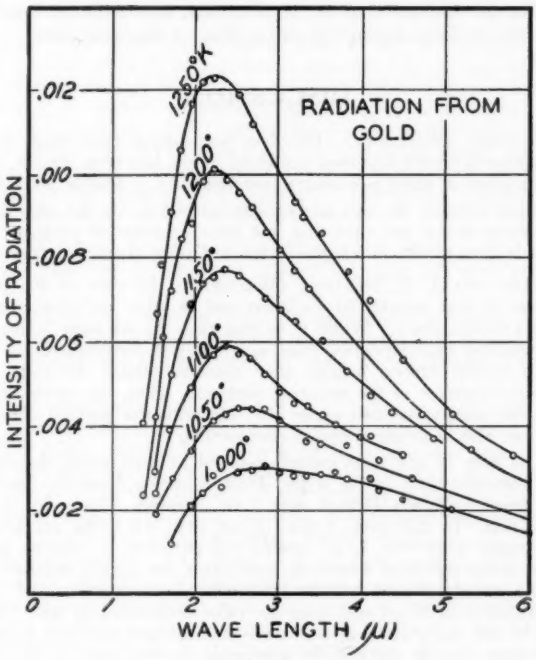

FIG. 12.

at 1353 K (Fig. 8) had a total emission of 2.5 times that of electrolytic cobalt-free nickel (Fig. 7) at the same temperature, and the energy for the wavelength of maximum emission was 2.8 times as great. This large difference was probably due to contamination of the surface by the ash contained in the impure specimen.

On checking the radiation equation of Aschkinass by using the set of distribution curves of the different metals at 1500 K (Fig. 2), it was found that this relation applied only very roughly to a few of the noble metals of the platinum series, while for most other metals it gave results which were not even approximately true.

XUM

ACKNOWLEDGMENTS

The author is indebted to Prof. C. E. Mendenhall of the University of Wisconsin for many valuable suggestions and for the apparatus which his laboratory furnished; to the United States Bureau of Mines for pyrometric and furnace equipment used in checking the transmission of color screens; and to H. L. Dunham for his aid in the computation of many of the corrections which were applied to the data.

Furthermore, the experimental part of the work could not have been carried out except for the assistance of M. H. Petersen, who adjusted instruments and took bolometric readings during the entire time of obtaining data.

DISCUSSION

W. E. FORSYTHE (WRITTEN): The data would have been much more valuable for some purposes if results had been expressed in absolute units, that is, if the energy flux had been given in watts per square centimeter for a definite wave length range.

The difference between the two sets of data on nickel, one for pure and the other for impure nickel shows the care that one must exercise in studying metals since Dr. Greenslade showed the difference in the radiating characteristics of nickel.

A. P. KRATZ AND A. E. HERSHEY (WRITTEN): In view of the present widespread interest in heat transfer by radiation and in other radiation phenomena, this paper is particularly timely. While it is true that the engineer is more concerned with total radiation characteristics than with spectral distribution, the results presented are of special interest because they show so plainly the wide variations of emissivity with character of the radiating surfaces. Thus, the curves in Figs. 7 and 8 show that the energy radiated varies not only with the physical characteristics of the surface, but also with the chemical composition.

That this is true of the total energy radiated as well as of the distribution of energy is commented upon in the paper, Heat Emission from the Surfaces of Cast Iron and Copper Cylinders Heated with Low Pressure Steam, by A. C. Willard and A. P. Kratz. In this case, it was shown that, while the surface of the new commercial copper plate was, to all intents and purposes, a "slightly polished" surface, the use of the published emissivity coefficients for slightly polished or calorized copper gave a calculated heat transfer by conduction, convection, and radiation that differed by as much as 35 per cent from the value determined by test. This deviation was caused by the uncertainty in the exact physical and chemical characteristics of the surface used, and is particularly interesting in the light of the facts brought out in Mr. Greenslade's paper.

The exact chemical and physical nature of a radiating surface must be known when selecting emissivity coefficients for radiant heat transfer calculations.

Since an arbitrary scale has been used for the energy measurements, it is very difficult to determine spectral emissivity coefficients from the published results. M. Cauley (Astrophysical Journal, Vol. 37, 1913, p. 164) shows that by making a brightness temperature determination at one wave length, it is possible to transform a spectral distribution for black body radiation to any arbitrary energy scale, and the question naturally arises as to whether or not this was done for the energy distribution curves presented in this paper. We feel that some such procedure would have greatly enhanced the value of the paper from the standpoint of the practical application of the results.

MR. GREGG: I would like to ask Dr. Greenslade if he investigated the effect of angle of emission on intensity of radiation. A recent German investigator, E.

Schmidt, from Maxwell's equations for radiation, calculated that the emission perpendicular from the surface of bright metals does not hold for oxidized surfaces, was less than that given by the co-sine, and if you wanted to get the total emissivity for all angles, you multiplied emissivity as determined by intensity from the perpendicular by four-thirds. Later investigators said that he made an unallowable assumption in arriving at this conclusion.

Schmidt made some experimental investigations and thought he proved that the intensity of emission as determined in a perpendicular direction should be multiplied by four-thirds and he also cited the fact that the edges of the metallic wire looked brighter than the center, which would follow from his conclusions that the radiation did not follow the co-sine.

F. B. ROWLEY: Dr. Greenslade is getting down to the fundamentals of heat transmission. In the past we have been prone to take test data for heat transmission in which there is a mixture of factors, and when we get through we know the combined effects but we haven't separated such factors as radiation, convection, and conduction. Dr. Greenslade might have included some of the materials which are being worked upon so intensively today, such as, aluminum, copper, and brass. His work has been at a higher temperature range than that encountered in the transmission of heat through building or refrigerator construction. Nevertheless, data in this range are of extreme value to the heating engineer.

PROF. A. C. WILLARD: This is the finest example of pure research which, in my experience with this Society, has been presented by any one at a meeting of the Society.

Dr. Greenslade has been asked to make the paper more practical. I myself have from time to time urged the importance of getting facts that were usable for this Society, and this gives me an opportunity to stress the conspicuous exception which proves the rule.

I believe there are certain fields in which we are vitally interested, from which advancement in our practice can only be obtained by such work as Dr. Greenslade has done and is doing. He has in a sense lifted a screen. You have looked beyond the practical field and you may say, "What use is it?" Why, gentlemen, it has all the use in the world. Unless we analyze with more precision and more exactness these fundamental things that are so essential to a complete understanding of the problem, we are never going to get anywhere fundamentally.

With the man interested in the field of pure research working with the man who is interested in the application of those results and their interpretation by the practicing engineer, it may be expected that any art would be advanced to that point where it can be looked upon as a science. The art of heating and ventilating still has a long road to travel before it becomes a science, but with the assistance of men like Dr. Greenslade, Mr. President, I think we can travel along that road with measurable speed and with a very definite possibility of ultimate success in placing this Society among the highly scientific bodies of the world.

PRESIDENT L. A. HARDING: I think engineers are convinced that we must continue to look to the physicist to point the way in much of our fundamental research. The difficulty is to get the physicist interested in our practical problems. A number of years ago when Professor Allen was Director of our Research Laboratory, he attempted to separate radiation and convection from the ordinary standard type radiator to determine some coefficients which could be applied to the Steffan-Boltzman law and, as I remember, he had considerable difficulty in maintaining a high vacuum in this chamber. After his results were out several years they were quite severely criticized on account of the fact that he did not maintain the correct vacuum, from the standpoint of the physicist. I would like to ask Dr. Greenslade what type of air pump was employed in these experiments.

DR. GREENSLADE: I used 4 pumps on the apparatus. Two pumps were for the high vacuum chamber and were placed in series. The highest vacuum pump was of the rotary molecular type. The molecular pump works on the principle of giving to the molecules of air a component of velocity toward the exit so that by rotating at high speed with small clearance between the moving and stationary surfaces, the molecules are virtually whipped away. This, of course, has to be backed up by another pump which operates between larger differences of pressure to reduce atmospheric pressure down to a point at which the molecular pump can work.

On the large chamber I used 2 pumps in series, one of them was of the ordinary rotary vein, oil-immersed type which works very rapidly and the other one for cleaning up the residual air, was of the rotary mercury type in which the parts rotate through mercury trapping the air and forcing it beneath the surface to the outside. This type of pump works with a difference of pressure of only 2 cm of mercury so it has to be backed up by another pump.

In regard to the radiation from the surface of the specimen, the flat surface of the radiating strip faced directly toward the receiving apparatus.

Concerning the changing of the arbitrary scale of intensity to absolute intensity, this can be done without the necessity of repeating the other work. It is accomplished by means of a small vacuum chamber with two windows, one window facing the flat surface of the specimen and one facing the black body "V." Then with the filament maintained at a certain definite temperature, determined by an optical pyrometer sighted into the "V," the black body temperature of the specimen is measured by another optical pyrometer sighted at the flat surface. Using the distribution equation it is now possible to solve for the constant of proportionality required for converting the corresponding radiation curve to an absolute intensity basis.

No. 894

INSULATING EFFECT OF SUCCESSIVE AIR SPACES BOUNDED BY BRIGHT METALLIC SURFACES

By L. W. SCHAD,[1] EAST PITTSBURGH, PA.

NON-MEMBER

AIR becomes ideal heat insulation if convection by it and radiation through it are prevented or greatly reduced. If, for instance, bright metal reflecting screens be placed in a heat path through air,[2] the transfer both by radiation and convection should be reduced. This paper contains information relative to the value of such screens in reducing thermal losses.

Two quite different test methods were employed for the purpose of obtaining data on the thermal transmission of walls built up of air spaces separated by bright metallic screens. In addition, some work was done to determine the convection and radiation constants of bright tin plate, the metal used for screens in this investigation. With the numerical values of these constants substituted in the usual heat transfer equations it is possible *first* to check the results obtained by direct test, and *second* to determine the amounts of energy transferred by conduction, convection and radiation respectively.

BOX TEST

Figs. 1, 2 and 3 show the construction of the box, which was tested both as a refrigerator and as an oven. Inside and outside surface temperatures were obtained with thermocouples soldered to the surfaces. When tested as a refrigerator the heat transfer was determined by the rate at which ice melted inside the box, and when converted into an oven heat was supplied electrically.

To facilitate comparisons with other heat insulating structures an overall heat transfer coefficient (k_t) is introduced which may be defined by the following well known equation for the conductivity of homogeneous solids bounded by plane parallel faces kept at constant temperatures t_1 and t_2, respectively:

[1] Research engineer, Westinghouse Electric and Manufacturing Company.

[2] Warmeschutz durch Aluminiumfolie, Prof. Dr.-Ing Ernst Schmidt, V. D. I. 71, 1395, Oct., 1927.

Presented at the 37th Annual Meeting of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, Pittsburgh, Pa., January, 1931.

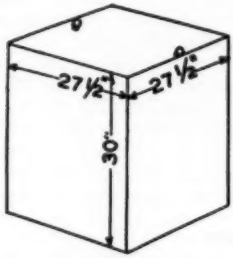

FIG. 1. ISOMETRIC VIEW OF EXTERIOR OF BOX TESTED SHOWING DIMENSIONS

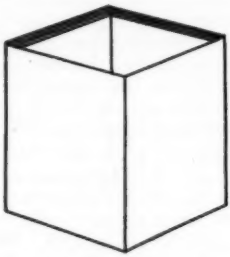

FIG. 2. ISOMETRIC VIEW OF BOX TESTED WITH LID OFF

$$*k_t = \frac{QL}{(A)\ (time)\ (t_1 - t_2)} \qquad (1)$$

where

k_t = Btu per hour per square foot at a temperature gradient of one degree Fahrenheit per inch

Q = total quantity of heat transferred

L = length of heat path

A = area of heat path

No rigid mathematical solution has been obtained for the shape factor of cubical enclosures surrounded by walls of uniform thickness. An approximate formula developed by Langmuir, Adams and Meikle,[2] applicable to the box in this test is:

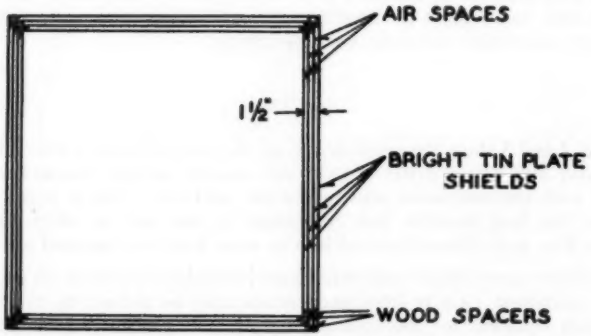

FIG. 3. SECTION THROUGH BOX TESTED, SHOWING TIN PLATE SHIELDS AND WOOD SPACERS PROVIDING FOR THE AIR SPACES

* As will be shown, k_t is not a constant, but is a function of t_1 and t_2. This should be taken into account when making comparisons.

[2] Dictionary of Applied Physics 1,463.

$$S = \frac{A}{L} + 0.54 \Sigma D + 1.2L \qquad (2)$$

where

S = shape factor
A = inside surface area
L = wall thickness
ΣD = summation of internal edges

Using the equation

$$k_t = \frac{Q}{(t_1 - t_2)\ (time)\ S} \qquad (3)$$

for the value of k_t the following results were obtained:

t_1	t_2	k_t
69 F	50 F	0.24
98	78	0.30
112	83	0.32
165	92	0.35

No great accuracy is claimed for the values of k_t given in the foregoing tabulation since it was impossible to maintain isothermal conditions on either the internal or external surfaces.

STANDARD TEST

The set-up for the usual method of guarded hot plate sandwiched between similar samples with guard ring along the edges is shown in Figs. 4 and 5. In this case three shields of bright tin plate were used in each sample and the air spaces were again ½ in. thick. Wooden frames separated the guard ring from the test samples and closed the outside edges of the guard ring. Ten-mil mica strips turned edgewise separated the plates. The following results were thus obtained:

t_1	t_2	k_t
176 F	103 F	0.40

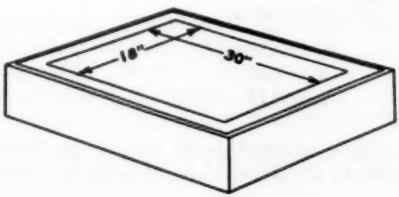

FIG. 4. PERSPECTIVE OF GUARDED HOT PLATE TEST APPARATUS

While it was possible to obtain fairly uniform temperatures over the hot and cold surfaces, there was undoubtedly some heat transmitted by conduction along the mica spacers and wooden frames so that the value of 0.4 for k_t is perhaps slightly high.

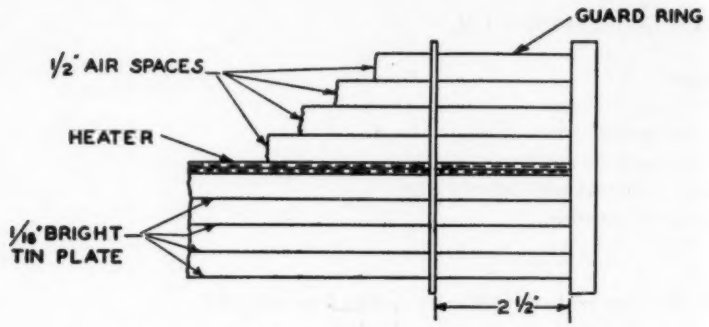

FIG. 5. SECTION THROUGH GUARDED HOT PLATE APPARATUS

HEAT DISSIPATION FROM BRIGHT TIN SURFACES

The usual equation for heat dissipation from surfaces by natural convection and radiation is

$$q = \frac{Q}{(A)\ (time)} = k_s\ (T_1 - T_2)^d + (e)\ (0.172) \left[\left(\frac{T_1}{100}\right)^4 - \left(\frac{T_2}{100}\right)^4\right] \qquad (4)$$

where

q = rate of heat transfer in Btu per hour per square foot
e = the radiation constant which is equal to unity for a *black body*
T_1 and T_2 are temperatures in degrees Fahrenheit absolute of the dissipating surface and surroundings respectively.

k_s and d are convection constants

The radiation and convection constants for a sheet of bright tin plate suspended in air were obtained from a curve of total heat dissipated plotted against surface temperature rise. It was necessary of course to maintain the surroundings at a fairly uniform temperature which in this work was 77 F. The numerical values of these constants were found to be:

$$k_s = 0.329$$
$$d = 1.25$$
$$e = 0.15$$

The convection constants will be modified as the screens are brought closer together, but they are applicable to air spaces ½ in. or more in thickness.[4]

THE EFFECT OF METALLIC SCREENS

Each of the methods of heat transfer—radiation, convection and conduction—will be considered separately as though the others were for the time being

[4] Dictionary of Applied Physics 1,474 and 475.

inoperative. The synthesis of the separate results in obtaining an overall coefficient of heat transfer introduces some error, which, however, is negligible if the temperature difference between the hot and cold plates is not over 200 or 300 F.

1. *Radiation:*

Consider two parallel surfaces of infinite extent, A the radiator and B the receiver. Let the temperatures of the surfaces be maintained at T_1 and T_2 degrees Fahrenheit absolute, respectively. The rate of heat interchange per square foot by radiation from A to B is

$$q = 0.172 \frac{e_1 e_2}{e_1 + e_2 - e_1 e_2} \left[\left(\frac{T_1}{100} \right)^4 - \left(\frac{T_2}{100} \right)^4 \right] \tag{5}$$

If both surfaces are alike, then $e_1 = e_2$ and

$$\frac{e_1 e_2}{e_1 + e_2 - e_1 e_2} = \frac{e^2}{2e - e^2} \tag{6}$$

Let a certain number of screens (n) be placed between the two surfaces and parallel with them. Now the heat flow rate from A to B is less than with no screens. If all surfaces are alike the problem is simplified and the effect of the screens can be predicted.

$$\text{Let } a = \left(\frac{T_1}{100} \right)^4$$

$$X_1 = \left(\frac{T_{s1}}{100} \right)^4$$

$$X_2 = \left(\frac{T_{s2}}{100} \right)^4$$

$$X_n = \left(\frac{T_{sn}}{100} \right)^4$$

$$b = \left(\frac{T_2}{100} \right)^4$$

where

T_1 = temperature of the hot or emitting surface A
T_{s1} = temperature of first screen
T_{s2} = temperature of second screen
T_{sn} = temperature of n^{th} screen
T_2 = temperature of the cool or receiving surface B

Let $\frac{0.172\, e^2}{2e - e^2} = E$, where E is a constant for all the screens as well as A and B if all surfaces are alike.

Let q_n = rate of heat transfer from A to B with n screens between the two surfaces. *Then*

$$\frac{q_n}{E} = a - X_1 = X_1 - X_2 = \ldots\ldots\ldots\ldots X_n - b$$

$$X_1 = a - \frac{q_n}{E}$$

$$X_2 = a - \frac{2q_n}{E}$$

$$X_n = a - \frac{nq_n}{E}$$

$$b = a - \frac{(n+1)\,q_n}{E}$$

$$q_n = \frac{(a-b)\,E}{n+1} = \frac{\dfrac{0.172\,e^2}{2e-e^2}\left[\left(\dfrac{T_1}{100}\right)^4 - \left(\dfrac{T_2}{100}\right)^4\right]}{n+1} \qquad (7)$$

That is, for n screens all having the same coefficient of emissivity as the emitting and receiving surfaces, the total radiation is $\frac{1}{n+1}$ times the total radiation without screens.

It is interesting to note that the temperature of any of the screens may be obtained also. Let it be required to find the temperature of the p^{th} screen where the total number of screens is n.

$$X_p = a - p\left(\frac{q_n}{E}\right)$$

$$but\ \frac{q_n}{E} = \frac{a-b}{n+1}$$

$$hence\ X_p = a - p\left(\frac{a-b}{n+1}\right)$$

and T degrees Fahrenheit (absolute) of the p^{th} screen is

$$100\sqrt[4]{a - p\left(\frac{a-b}{n+1}\right)}$$

2. Convection:

Free convection in air is given by the equation

$$q = C\,(t_1 - t_2)^d \qquad (8)$$

C and d are constants to be determined experimentally.

$(t_1 - t_2)$ is the average temperature difference between the heated surface and ambient air.

The heat transfer by convection across an air space bounded by two parallel surfaces is $q = \frac{C}{2^d}\,(t_a - t_b)^d$ in which $(t_a - t_b)$ now represent the average temperature difference between the two surfaces.

Let $k_a = \frac{C}{2^d}$ and the convection equation becomes

$$q = k_a\,(t_a - t_b)^d \qquad (9)$$

With n screens between the two surfaces

$$q_n = k_a (t_a - t_1)^d = k_a (t_1 - t_2)^d = q_n = k_a (t_a - t_1)^d = k_a (t_1 - t_2)^d = k_a (t_n - t_b)^d$$

where q_n = rate of heat transfer between surfaces A and B with n parallel surfaces equally spaced between them

t_a = temperature of surface A.

t_1 = temperature of first screen.

t_n = temperature of n^{th} screen.

t_b = temperature of surface B.

Let $(t_a - t_1)$, $(t_1 - t_2) \ldots . (t_n - t_{n-1})$, $(t_n - t_b) = X_1, X_2 \ldots . X_n$ and X_{n+1}

Let X_t = total temperature drop from surface A to surface B. Since $k_a X_1^d = k_a X_2^d$, etc.

$X_1 = X_2 = \ldots . . . . . X_n = X_{n+1}$, and

$X_t = (n+1) X_1 = (n+1) X_2$, *or*

$$X_1 = \frac{X_t}{n+1}, \text{ and}$$

$$q_n = \frac{k_a (X_t)^d}{(n+1)^d} \qquad (10)$$

3. *Conduction:*

With n screens per inch of heat path each screen L inches thick

$$q_n = \frac{t_1 - t_2}{\frac{1 - nL}{k_a} + \frac{nL}{k_m}} \qquad (11)$$

$t_1 - t_2$ = temperature difference between hot and cold surfaces.

k_a = thermal conductivity of air.

k_m = thermal conductivity of screens.

It is apparent therefore that the amount of heat transferred by conduction is increased by the use of metal screens, but of course the thinner the screens the less this increase will be.

SUMMARIZING AND COMPARING

The equation for the rate of heat transfer across an air space without screens is

$$q = \frac{(T_1 - T_2) k_a}{L_a} + k_a (T_1 - T_2)^d + \frac{0.172\, e^2}{2e - e^2} \left[\left(\frac{T_1}{100} \right)^4 - \left(\frac{T_2}{100} \right)^4 \right] \qquad (12)$$

The corresponding equation for the case where n screens are used is obtained by summing equations (7), (10) and (11) and is as follows:

$$q_n = \frac{(T_1 - T_2)}{\frac{L_a}{k_a} + \frac{L_m}{k_m}} + \frac{k_a (T_1 - T_2)^d}{(n+1)^d} + \frac{0.172\, e^2 \left[\left(\frac{T_1}{100} \right)^4 - \left(\frac{T_2}{100} \right)^4 \right]}{2e - e^2} \qquad (13)$$

In equations (12) and (13)

q = rate of heat transfer without screens.

q_n = rate of heat transfer with n screens.

T_1 and T_2 = temperatures in degrees Fahrenheit (absolute) of hot and cold surfaces.

L_a = length of heat path through air.

L_m = length of heat path through metal screens.

k_a = thermal conductivity of air.

k_m = thermal conductivity of metal used for screens.

k_s and d = convection constants, determined experimentally.

e = emissivity of metal used in per cent of black body radiation.

HEAT TRANSFER THROUGH AIR SPACES

The value of air spaces as thermal insulation is often discussed without reference to the character of inclosing walls. The character of bounding surfaces greatly affects the amount of energy transferred by radiation which is the only way of modifying the total heat transferred across a given air space. The importance of radiation is often overlooked, especially in problems involving ordinary room temperatures. To show the relative effects of radiation, convection and conduction it is only necessary to evaluate equations (12) and (13) with the numerical values of the various constants inserted.

The following examples will illustrate the use of these equations and at the same time show how much the transfer across a given air space may be modified. The distance between the hot and cold walls is 1½ in. and the temperatures of the hot and cold surfaces are 212 F and 32 F, respectively. In *Case 1* the inclosing walls are paper, wood, asbestos or other similar material having a radiation constant of approximately 0.9, while in *Case 2* the walls are bright metal with a radiation coefficient of 0.15. In *Case 3* two bright metal screens are used thus dividing the inclosure into three ½-in. spaces.

Case 1

k_a, or air conductivity at 122 F is 0.174

k_s, convection constant = 0.141

d, convection constant = 1.25

$\frac{e^2}{2e - e^2}$ radiation constant = 0.82

Substituting these values in equation (12)

$$q = \frac{(0.174)\ (180)}{1.5} + 0.141\ (180)^{1.25} + (0.172)\ (0.82)\ [(6.72)^4 - (4.92)^4]$$

*$q = 21 + 92 + 206 = 319$

* Computations by slide rule and values given to nearest whole number.

The heat flow in Btu per square foot per hour is therefore as follows:

21 Btu or 6.6 per cent of the total is transferred by conduction.
92 Btu or 28.9 per cent of the total is transferred by convection.
206 Btu or 64.5 per cent of the total is transferred by radiation.

319 total Btu per square foot per hour.

The overall transfer coefficient is

$$k_t = \frac{(319)\ (1.5)}{180} = 2.66$$

Case 2

The only change in *Case 2* is the radiation coefficient which instead of 0.82 as for *Case 1* is 0.081. Substituting and solving as before the following results are obtained:

21 Btu or 15.8 per cent of the total is transferred by conduction.
92 Btu or 69.1 per cent of the total is transferred by convection.
20 Btu or 15.1 per cent of the total is transferred by radiation.

133 total Btu per square foot per hour.

The overall coefficient of heat transfer is

$$k_t = \frac{(133)\ (1.5)}{180} = 1.11$$

By replacing the surfaces in *Case 1* with bright metal the heat flow has been reduced to less than one-half of the previous value. This was accomplished entirely by reducing the radiation, without changing the convection and conduction.

Case 3

As already stated, the conditions for *Case 3* are the same as for *Case 2* except that two bright metal screens now divide the inclosure into three ½-in. spaces. Solving equation (13) with the numerical values of the constants substituted and assuming a screen thickness of 1/16 in. the following is obtained:

$$q_n = \frac{180}{\dfrac{1.5-2(1/16)}{0.174} + \dfrac{0.125}{273}} + \frac{\text{previous convection}}{(2+1)^4} + \frac{\text{radiation of } \textit{Case 2}}{2+1}$$

$= 23 + 23 + 7 = 53$ which means that in one hour and across one square foot of surface

23 Btu or 43.4 per cent of the total is transferred by conduction
23 Btu or 43.4 per cent of the total is transferred by convection
7 Btu or 13.2 per cent of the total is transferred by radiation

53 total Btu per square foot per hour.

The overall coefficient of heat transfer is

$$k_t = \frac{(53)\ (1.5)}{180} = 0.44$$

DISCUSSION OF RESULTS

The coefficient of heat transfer as derived from the direct tests averages about 0.38 at an average temperature of 135 F while the corresponding value obtained by the indirect method is 0.44 which is quite satisfactory agreement in view of the uncertainties involved.

As stated previously some error is introduced by considering the three different ways of heat transfer separately in obtaining the rather *synthetic* coefficient. The writer knows no method of obtaining an expression for the heat transfer coefficient through successive air spaces by using the radiation, convection and conduction constants simultaneously except by trial and error which would also be more or less approximate and very much more laborious.

SUMMARY

The rate of heat flow between two parallel surfaces, A and B, in air separated a distance L from each other is given by the equation

$$q = \frac{k_a\ (T_1 - T_2)}{L} + k_s\ (T_1 - T_2)^d + \frac{(e_1 e_2)S}{e_1 + e_2 - e_1 e_2}\left[\ (T_1)^4 - (T_2)^4\ \right] \quad (14)$$

where

k_a = thermal conductivity of air

k_s and d = convection constants

e_1 and e_2 = the radiation constants of the respective surfaces

S = black body emissivity = 0.172×10^{-8}

T_1 and T_2 = the absolute temperatures of the respective surfaces

q = Btu per square foot per hour

k_s = approximately 0.14 for most smooth flat surfaces

d = 1.25

e = approximately 0.9 for wood, paper, asbestos, etc., and from 0.05 to 0.20 for bright metallic surfaces.

Where bright metal screens are used between two surfaces of like material, that is for the case of successive air spaces between parallel surfaces of bright metal, the rate of heat transfer is given by the equation

$$q_n = \frac{T_1 - T_2}{\frac{L_a}{K_a} + \frac{nL_m}{K_m} + \frac{k_s\ (T_1 - T_2)^d}{(n+1)^d} + \frac{e^2 S}{\frac{2e - e^2}{n+1}(T_1^4 - T_2^4)}}$$

where

q_n = rate of heat transfer from surface A to surface B with n screens dividing the inclosed space from A to B in $n+1$ equal parts.

L_a = length of heat flow path in air.

k_a = thermal conductivity of air.

L_m = thickness of metal screen.

k_m = thermal conductivity of air.

With the proper value of the constants substituted in the foregoing equations it is not only possible to determine q and q_n but also the amounts of heat transferred by and therefore the relative importance of conduction, convection and radiation in problems of heat flow across air spaces.

Three examples involving the use of the foregoing equations have been given which are briefly summarized as follows:

Case 1.

Air space without screens between two good radiating surfaces such as cork, paper, wood, etc. Let $e = 0.9$, which is approximately correct for such materials.

Case 2.

Air space without screens between two poor radiating surfaces such as bright metals where $e = 0.15$.

Case 3.

Same conditions as for *Case 2*, with the exception that two bright tin plates divide the 1½-in. air space into three ½-in. spaces. Common to all three cases: distance from the hot to the cold surfaces is 1½ in. and the respective temperatures are 212 F and 32 F.

A summary of the results is given in Tables 1 and 2.

TABLE 1. BTU TRANSFERRED PER HOUR PER SQUARE FOOT BY CONDUCTION, CONVECTION AND RADIATION

	Case 1	Case 2	Case 3
Conduction	21	21	23
Convection	92	92	23
Radiation	206	20	7
Total	319	133	53

TABLE 2. PERCENTAGES OF TOTAL HEAT TRANSFERRED BY CONDUCTION, CONVECTION AND RADIATION AND VALUES OF OVERALL HEAT TRANSFER COEFFICIENT

	Case 1	Case 2	Case 3
Conduction	6.6%	15.8%	43.4%
Convection	28.8%	69.1%	43.4%
Radiation	64.5%	15.1%	13.2%
Coefficient (k_1)	2.66	1.11	0.44

EFFECT OF BRIGHT METALLIC SCREENS ON CONDUCTION, CONVECTION AND RADIATION

The transfer by conduction is increased, the amount of increase depending upon the extent to which the air path, a poor conductor, is displaced by metal, which is a much better conductor. Convection is decreased as follows: The heat transfer by convection is given by

$$q = K_S (t_1 - t_2)^d \text{ without screens.}$$

$$q = K_S \frac{(t_1 - t_2)^d}{(n+1)^d} \text{ with } n \text{ screens.}$$

Radiation is decreased, the amount of decrease being conveniently shown as follows:

Let q = transfer by radiation without screens.

Then $\frac{q}{n+1}$ = transfer by radiation with n screens.

Experiments

Two direct tests were made to determine the heat transfer coefficients of successive half-inch air spaces separated by bright tin plate.

Standard Test

The usual method of guarded hot plate sandwiched between two similar samples with guard ring along the edges gave a transfer coefficient of 0.40 Btu per square foot per hour per degree Fahrenheit per inch with t_1 and t_2 at 176 F and 103 F, respectively.

Box Test

A box having inside dimensions of 24 × 24 × 26 in. was tested both as a refrigerator and as an oven. The walls of the box were 1.75 in. thick with three ½-in. air spaces separated and inclosed by 1/16-in. bright tin plate. The following results were obtained:

t_1	t_2	k_1
69 F	50 F	0.24
98	78	0.30
112	83	0.32
165	92	0.35

DISCUSSION

J. L. GREGG (WRITTEN): The Battelle Memorial Institute's data on the heat conductivity of air layers formed by bright metallic surfaces agree generally with conductivity values obtained by Mr. Schad. In the majority of cases we have used bright aluminum foil as separators instead of tin-plate.

While Mr. Schad's calculations of the heat transferred by radiation and convection appear to be exact and presented in a unique and useful form, he undoubtedly has erred in calculating the quantity of heat transferred by convection in air layers ½ in. thick. E. Schmidt [1] [2] and Mull and Reiher [3] have shown that in air layers approximately ½ in. thick the quantity of heat transferred by convection is only a small fraction of the quantity transferred by conduction, while according to Mr. Schad's analysis the quantity of heat transferred by convection through a ½-in. air layer when the difference in temperature between the opposite faces is 60 F is equal to the quantity transferred by conduction.

[1] Schmidt, E., Wärmeschutz durch Aluminiumfolie, Zeit. V.D.I., 71, 1927, pp. 1395-1400.
[2] Schmidt, E., Versuche über den Wärmeübergang in ruhender Luft, Zeit. f. d. ges. Kälte-Industries, 35, 1928, pp. 213-219.
[3] Mull, W. and Reiher, H., Der Wärmeschutz von Luftschichten, Beihefte zum Gesundheits-Ingenieur, Series 1, No. 28, 1930.

Data obtained at the Institute also indicate that the heat transfer by convection through ½-in. air layers is inappreciable. The tests were made on a guard ring hot plate 20 in. square, in which 2 hot plates instead of 2 samples are used. The apparatus is ordinarily used with the heat travelling up through the sample, but it can be turned over so that the heat travels down, or set vertically so that the heat traverses vertical air layers. It was found that in a unit composed of several ½-in. air layers bounded by bright aluminum foil the apparent heat conductivity did not vary with the position of the apparatus, which would indicate no appreciable heat transfer by convection, for if there were appreciable transfer by convection the conductivity values would be less when the hot plate was at the upper surface of the sample than in the other two cases.

Calculations based on the assumption of a negligible transfer of heat by convection in ½-in. air layers indicate exceptionally good insulating values for structures consisting of air layers formed by bright metallic surfaces. If Mr. Schad's calculated values for ½-in. air layers are changed by assuming no transfer of heat by convection the coefficient of heat transfer would be 0.25 Btu per hour per square foot per degree Fahrenheit per inch thickness instead of 0.44 Btu, while the value he determined experimentally is 0.38 Btu. The fact that this value is appreciably higher than the value calculated on the basis of no transfer of heat by convection may be due to the impossibility of maintaining isothermal conditions on either the internal or external surfaces, which was mentioned by Mr. Schad. Other studies at the Institute reveal that a structure consisting of a series of foil covered cardboard sheets held ½ in. apart by non-metallic separators has a coefficient of heat transfer of about 0.25 Btu, when the mean temperature is about 100 F and the temperature drop across 3 air layers is 45 F. This value is only slightly higher than the calculated value based on no transfer of heat by convection, and indicates that convection losses are negligible.

E. R. QUEER[4] (Written): From a historical standpoint it is interesting to note that in 1915 Dickinson and Van Dusen of the Bureau of Standards presented a paper[5] before the American Society of Refrigerating Engineers in which they pointed out the insulating effect of bright metal surfaces bounding air spaces. In this paper they separated the heat transferred by radiation from the total heat flowing, the remaining amount being combined as conduction and convection. Later, P. Nicholls presented a discussion on this paper.[6]

We have been interested in the insulation possibilities of aluminum foil for some time and some results of tests made to check the validity of equation No. 13 of Mr. Schad's paper are given in Table A.

TABLE A—RESULTS OF TESTS CHECKING VALIDITY OF SCHAD'S EQUATION NO. 13

Heat Input Measured	Heat Input Calculated from Eq. 13	High Side Temp. F	Low Side Temp. F	Mean Temp. F	Conductance Btu/Sq Ft/ΔT Test Value	Per cent Increase in Calculated Heat Input
9.90	14.66	82.4	44.1	63.1	0.259	48.1
22.23	30.15	129.5	55.0	92.3	0.299	35.6
40.25	51.53	189.7	71.5	130.6	0.340	28.0

Two ½-in. air spaces on either side of hot plate. Hot and cold surfaces lamp blacked—screens of aluminum sheets 0.003 in. thick. Combined emission factor = 0.10.

In these tests the setup was virtually the same as that of the author except that the hot plate is a 12 x 12-in. plate having a 9 x 9-in. test area and both 0.003 and

[4] Instructor in Engineering Research, Pennsylvania State College.
[5] Published in A. S. R. E. Journal, Sept., 1916.
[6] Published in A. S. H. & V. E. Journal, Nov., 1921.

0.004-in. rolled aluminum screens were used for the bright surfaces. A constant voltage was placed across the hot plate and after thermal equilibrium was reached the temperatures and temperature differences of the hot and cold plates were substituted in the author's equation.

The agreement between test and calculated values became progressively better in going from ¼ to ½-in. and then to 1-in. air spaces. In a vertical 1-in. air space with black surfaces on both the hot and cold plates the agreement was good.

The difficulty in using the equation arises in the author's attempt to separate the heat transferred by conduction and that transferred by convection. These two items are a process of diffusivity in an enclosed air space and are too closely allied to differentiate between them. As heat is emitted from a surface bounded by air, part goes by radiation, the remainder by conduction until the stationary air film on the surface is passed, and from there on mostly by convection. The same applying in a reverse order for a receiving surface. In the paper the heat transfer by conduction and convection are separately applied to the full width of the enclosed air spaces. This must be modified before the equation is acceptable.

The part of the equation dealing with radiation is accepted as standard and has conclusive data to substantiate it. However, as pointed out previously by Willard and Kratz, there is great need for more data on the emissivity factor.

The author is to be commended for his work on this problem. It is hoped more data will be presented in the near future relating to this phase of thermal insulation, as it is believed to have a good field of application in refrigeration, possibly house insulation, and in ovens operating at temperatures up to 350 F.

PRESIDENT HARDING: Engineers, particularly mechanical engineers, I believe, have always been more or less radiation-minded. Those of you who are familiar with the history of the development of the steam engine will recall that it was the practice of James Watt to lag steam cylinders in order to reduce the heat emissivity and consequently the condensation loss. Shortly thereafter it became the practice to cover the lagging with a polished iron or brass surface. So it is not a particularly new subject, this radiation proposition.

P. NICHOLLS: About 1830 Peclet developed general equations for multiple air spaces for walls, envelopes and windows. This was before the fourth power law for radiation had been discovered, and indeed it is not necessary for low temperatures and small differences. Such references to historic work should be an incentive to us; and I suggest that the author will not feel proud of his results when comparing them with those of 100 years ago, because Peclet states his for a certain construction, as follows:

Number of air spaces.....	0	1	2	3	4
Relative transmission, computed ...	1	.87	.77	.69	.62
Relative transmission, by test.....	1	.90	.75	.67	.60

The derivations of the formulae by the author are timely because I understand that air space transmission is being studied at a number of institutions at present. However, I suggest that it is fundamentally incorrect to use a general formulae including both conduction and convection, the constants for which are presumedly to be derived independently. In fact convection laws and values derived experimentally include conduction and the two can not be separated except arbitrarily by deducting a computed conduction; this, however, would not correspond to the transfer action because the temperature gradient assumed for the conduction is destroyed by the movement of the air.

It is true that the transfer other than by radiation tends to become all conduction as the thickness of the space gets small, and that probably one could fit the author's general equation to experimental results if one introduced variable functions of the

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thickness into the conduction and convection factors, but it is simpler to consider them as one and to derive a single space conduction-convection factor dependent on the thickness.

Peclet recognized this and deduced by computation that the optimum thickness of the space was about ¾ in., a fact that has been confirmed by all investigators since then.

I understand that about 6 institutions are investigating air spaces at this time; it seems a pity that their efforts can not be combined and the work done thoroughly and be carried to completion.

J. D. EDWARDS: This paper is a timely contribution to the subject which is being given active consideration by both users and manufacturers of insulating materials. Although Mr. Schad's investigations are limited to the use of tin plate for the bright metallic surfaces used to confine the air spaces, obviously the same principles apply if other metals are employed, although the effective constants will be somewhat different. There has been a very substantial application of metal foil seal insulation in Germany where they have employed aluminum foil for the bright metallic surfaces. This type of insulation has been known as "alfol." Commercial development in the United States is also looking toward the use of aluminum foil for this type of insulation.

Mr. Schad has not attempted to give a complete theoretical explanation of the phenomena involved, but has merely attempted to show how far the available theory will go towards explaining the observed facts. In doing that he has performed a very useful service.

With regard to aluminum foil insulation, various values have been given where the emissivity of the aluminum foil ranges from about 0.02 to 0.15. If we assume an emissivity value of 0.10 for aluminum foil and recalculate the values for Case 3, the heat transfer through similar aluminum foil streams one thousandth of an inch in thickness and 3½-in. air spaces by conduction, convection and radiation, will be 21, 23 and 5 Btu respectively as compared with 23, 23 and 5 Btu for the case where the bright tin plate separators were used. The difference is not great and, of course, the difference in the calculated values come from using a lower emissivity for the aluminum and the fact that in using aluminum foil a thinner separator is used.

A box test of the thermal insulating value of aluminum foil air cell insulation has been made by Taylor and Mason. The construction of the foil insulating panels was somewhat different than that employed by Mr. Schad. Nevertheless, for an average temperature of 60 the heat transfer coefficient varied from 0.25 to 0.28, for panels having ½-in. thickness air cells and 1 to 3 aluminum foil separators. These values are quite comparable with the value 0.24 lb by Mr. Schad.

From the standpoint of the user and manufacturer of this type of insulation, it is not a matter of grief that observed values are somewhat lower than those would have been predicted from theory.

Another point of practical importance in considering insulation of this type is that the low emissivity of the outer aluminum foil surface will be effective in increasing the resistance of heat transfer from the foil surface to the air and hence increases the efficiency of utilization of this type of insulation.

Mr. Schad is to be congratulated on his analysis of the factors determining the efficiency of this type insulation. It can be readily profited by all industries in metal foil air cell insulation.

PRESIDENT HARDING: I have asked Professor Wood, Past-President of the *American Society of Refrigerating Engineers,* to say a word in regard to this paper.

PROFESSOR WOOD: If this paper has done nothing else, it has brought out a discussion that I believe is going to lead toward a satisfactory solution some day of the old question of air spacing.

The work which is being undertaken in the Engineering Experiment Station at Pennsylvania State College, I hope, will supplement what has been done in other places. I am sure of the care with which it is being planned and executed, and the fact that apparently there is a difference in the results when tested by or lined up with equation No. 13 indicates there may be yet a possibility for a reconsideration of some of the constants.

I am not sure whether convection is some form of conduction, or conduction is some form of convection, and for all practical purposes that have come to my attention in 15 years, I don't care.

We should first make a clear-cut division between radiation and everything else. You can call it conduction, but let us make that division first, the same as Mr. Nichols did with the Bureau of Standards' results a good many years ago on air spaces, when he showed that approximately so much was due to radiation and the rest to what he called convection.

When air moves in a space, some of the heat, of course, is carried. How is it carried? Do you call it convection if it is carried? Yes, naturally, but at the same time there may be other heat rays that we have not accounted for in that flow of that particle of air from the one surface to the other, and you may call that conduction. I don't know. We need a little more analysis of the fundamentals.

PRESIDENT HARDING: Thank you, Professor Wood.

MR. PENNEY: I feel that there is an advantage in trying to separate convection and conduction. Perhaps it is an academic one but it seems to me, as I would design it, that the fundamental air conduction is the heat that is transferred by the molecular motion of the gases and that we would define convection as the heat that is carried due to air movement.

Of course, in order—before we get this air movement, we transfer heat from the surface to the gas, change its density, that is this heat is first transferred from the surface to the gas fairly close to it by molecular movement or by what we term conduction, and then this causes that air to be lighter and produces a circulation and I don't believe that Mr. Schad or any of us would claim that we have the final constants in getting this heat that is transferred by convection, because, obviously, this circulation of the air is going to depend on the separators we use between our surfaces, on the spacing and on lots of other complicated factors. I think it is quite a big advantage to show approximate calculations for that and probably as we do more research work on it we will find more exactly what this convection constant is. I also believe it is a real advantage to start to separate just the heat that is transferred from one surface to another by the molecular motion or by the conductivity of the gas and the heat that is transferred by the circulation of the gas or, as we usually term it, convection.

PRESIDENT HARDING: Mr. Schad, will you close the discussion?

L. W. SCHAD: This paper was intended to emphasize the effect of bright metal screens on the heat flow through air spaces rather than to give the convection constants to a high degree of accuracy for all conditions that might be encountered in such problems.

There is sufficient fundamental data given in the paper to furnish a fairly good estimate of heat transfer across air spaces met with in practical work.

Further work should be done along this line. It would be interesting and useful to determine convection as a function of all the possible modifying conditions such as thickness of air space, shape, size, and orientation of surfaces involved as well as temperature difference between surfaces.

No. 895

WIND VELOCITY GRADIENTS NEAR A SURFACE AND THEIR EFFECT ON FILM CONDUCTANCE

By F. C. HOUGHTEN [1] *(MEMBER)*, AND PAUL MCDERMOTT [2] *(NON-MEMBER)*,
PITTSBURGH, PA.

DURING the past few years considerable research has been directed towards a study of heat transfer coefficients and their application to heat flow through typical building constructions under natural conditions of weather.

As a result of these studies, students of heat transfer and designing engineers have greater confidence in the application of the formula [3] for the calculation of overall coefficients from coefficients for the component factors which go to make up the total resistance to heat flow through a wall. At the present time the only factors in the heat transmission equation which are not accepted with confidence are the film conductance coefficients f_1 and f_0 for inside and outside wall surfaces. Confidence is particularly lacking in the outside coefficients for different walls for various weather and wind conditions.

The reason for lack of confidence in the available surface conductance coefficients is not found in insurmountable difficulties encountered in determining these values for any particular condition, but rather in difficulties encountered in the application to practical conditions of available coefficients determined under some particular test arrangement. The most important factor affecting the application of the coefficients is the prevailing wind and its effect on air velocities near the wall surface.

A coefficient may be determined with great accuracy in the laboratory in a particular piece of apparatus where a uniform wind velocity is maintained parallel to, and measured at some definite distance away from, the surface. This value is reported as applying to a wind velocity equal to that measured in the experimental apparatus. There is no assurance of the application of this coefficient to a wall subjected to any particular wind velocity in open space.

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[1] Director, A. S. H. V. E. Research Laboratory.
[2] Research Engineer, A. S. H. V. E. Research Laboratory.
[3] See Chapter 3, A. S. H. V. E. GUIDE, 1931.

Presented at the 37th Annual Meeting of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, Pittsburgh, Pa., January, 1931, by F. C. Houghten.

In order to establish data on the effect of wind velocity gradients near a surface on film conductance, the A. S. H. V. E. Laboratory undertook a study of the subject in accordance with a plan outlined by the 1928 Technical Advisory

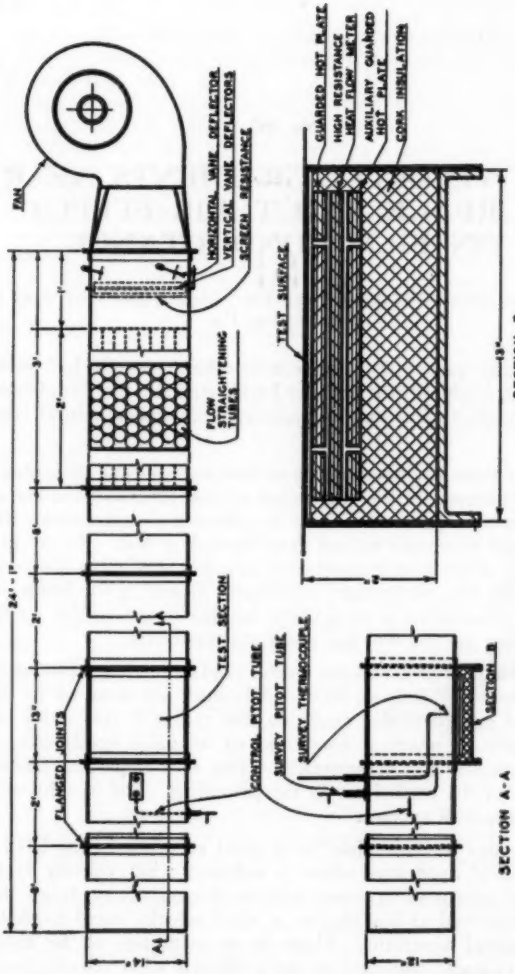

FIG. 1. APPARATUS FOR STUDYING EFFECT OF WIND VELOCITY GRADIENTS ON FILM CONDUCTANCES

Committee on Heat Transmission under the chairmanship of L. A. Harding. The problem as outlined included two rather distinct phases; *first*, a study of the effect of direction and location of measurement of wind velocity near a

surface on the film conductance coefficient; and *second*, a study of wind velocity gradients near a wall in free space and in a confining duct. This report includes data on both phases of this subject based on the direction of the wind parallel to the surface. It does not touch upon the more complicated problem of determining these effects when the wind impinges against a wall at some other angle.

DETERMINATION OF FILM CONDUCTANCE COEFFICIENTS

The apparatus used for studying the effect of wind velocity gradients and confining duct size on film conductance coefficients is shown in Fig. 1. The desired air velocities through the 24-ft duct, including the test section, were produced by the fan. Vane deflectors were placed in both the vertical and horizontal dimensions of the duct near the fan outlet. These were adjusted to give natural distribution of flow. A window screen resistance was placed just beyond the deflectors, to further aid in obtaining natural distribution. A short distance beyond this resistance the entire cross section of the duct was filled with sheet metal tubes, 2 in. in diameter and 2 ft in length, so placed that the air had to pass through them. This had the effect of straightening out any swirling motion of the air.

The duct was 12 in. x 14 in. with the 12-in. dimension horizontal and built with particular care so as to give smooth, even sides. The joints were flanged and particular care was taken to insure that the inside surfaces of the duct through the flanged joints were even. In order to determine the effect of the width of the duct perpendicular to the test surface on the film conductance coefficient, the temperature gradient, and the wind velocity gradient away from the surface, provision was made for dividing the duct into two equal parts by a vertical partition through the center. This made it possible to study, in both a 6-in. and a 12-in. duct, the phenomena described.

The test surface was set up in the enlarged test section of the duct in such a way that its surface was in the plane of one of the vertical sides. A thermocouple was arranged so that the temperature gradient away from the test surface across the greater part of the width of the duct could be surveyed. The copper constantan thermocouple, of No. 36 B and S wire, was held by two prongs of a fork so that its junction and one inch of the wire on either side of it were located in the isothermal plane. Pitot tubes were arranged so that the velocity gradients throughout both the horizontal and vertical dimensions of the duct could be surveyed. A control thermocouple and a control Pitot tube were provided in order to check any change in conditions during a survey.

The enlarged section *B*, Fig. 1, shows the arrangement of the surface and heater and the method of applying heat to the surface. The heater consisted of two standard guarded hot plates designated as *main* guarded hot plate and *auxiliary* guarded hot plate. These were separated by a high resistance heat flow meter. The heat input into the main guarded hot plate was adjusted so as to maintain the desired temperature of the test surface. The heat input into the auxiliary guarded hot plate was then adjusted so that no temperature difference or heat flow was indicated through the high resistance heat flow meter. Under these conditions all of the heat developed in the center of the main guarded hot plate necessarily flowed outward through the test surface.

The test surfaces were made as thin as possible and cemented directly to the copper face of the main guarded hot plate so as to offer as little resistance

as possible to the flow of heat from the main heater to the surface. The heaters were 12 in. square with ½-in. cork insulation around the edges. The test surface, 13 in. square, joined the 12 x 14-in. duct, at either end of the test section, in a smooth plane.

In making a test the surface studied was maintained at the desired temperature and the guard ring of the main heater, the auxiliary heater, and its guard ring were adjusted to the same temperature. These conditions were main-

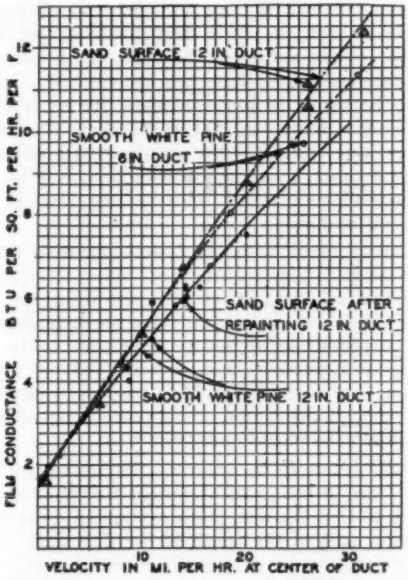

FIG. 2. FILM CONDUCTANCE COEFFICIENTS FOR SMOOTH PINE AND SANDED SURFACES AT VARIOUS AIR VELOCITIES

tained for a sufficient length of time to establish equilibrium of heat flow between the surface and the air. The temperature and velocity of the air through the duct were maintained constant. The heat dissipated from the surface was calculated from the electrical input supplied to the center section of the main heater. The film conductance coefficient was calculated from the formula

$$f = \frac{EIJ}{A\,(t_s - t_a)} = \frac{H}{t_s - t_a} \qquad (1)$$

where

I = current input in amperes.

E = emf across the effective part of the heater.

J = the equivalent of heat = 3.415 Btu per hour.

A = area of the heater winding or the center of the main guarded hot plate in square feet.

t_s = temperature of the test surface.

t_a = temperature of the air in the center of the duct.

H = heat input by the heater in Btu per square foot per hour.

f = the film conductance coefficient in Btu per square foot per hour per degree temperature difference between the surface and the air.

Tests were made for a large number of velocities ranging from 1 mph, the lowest which could be maintained uniformly, to a maximum of 35 miles per hour for the 6-in. duct, and 20 mph for the 12-in. duct.

The film conductance coefficient for the painted sanded surface in the 12-in. duct and for the smooth white pine surface in the 12-in. and 6-in. ducts are plotted against wind velocity in Fig. 2. The painted sanded surface was made by cementing paper directly to the copper face of the main heater, and then applying to the paper a single coat of light gray paint. While the paint was still soft, sand which was graded to pass through a 60 mesh sieve and over an 80 mesh sieve was sprinkled evenly over the surface so as to give a distribution of approximately one grain of sand per 1/16-in. center. After the first coat of paint into which the sand had been sprinkled was dry, the sanded surface was painted a second single coat. This left each grain of sand covered with a paint film in such a way that the shape and size of the grains stood out rather distinctly in the painted surface. This surface was taken to simulate a painted, sand finish plaster. After obtaining the values for this surface plotted in Fig. 2, the surface was painted an additional two coats of the same grade of paint. These coats of paint tended to fill up the spaces between the sand grains and to smooth off the surface so as to produce a fairly smooth surface with rounded humps. Three tests were made on this surface for a wind velocity of approximately 14 mph resulting in the points in squares showing a film conductance coefficient considerably lower than the values for the rough sanded surface. The paint used was mixed at the laboratory and consisted of 71 per cent of white lead ground in oil, 23 per cent white zinc ground in oil, 6 per cent boiled linseed oil, sufficient turpentine to give proper paint consistency and enough lamp black to give a light gray tinge.

The smooth white pine surface, made by planing and sanding a board down to a thickness of 1/16 in. and cementing it to the copper face of the main heater, was adjusted so that its exposed surface lay in the plane of the duct wall. The curves in Fig. 2 show the relationship between the film conductance coefficients for the smooth pine surface as measured in the 12-in. and 6-in. ducts.

DETERMINATION OF WIND VELOCITY GRADIENTS

There are very little data available in the literature on the subject of wind velocity gradients near a surface, and the little that is to be had seems to have been determined on very small surfaces which were in, or which formed part of the walls of a wind tunnel. A large part of the recorded work was done by aerodynamic engineers in the interest of aviation. Often the surface studied was cylindrical, or otherwise curved. Perhaps the best data available on the subject resulted from an investigation by the National Advisory Committee

for Aeronautics.[4] The purpose of this study was to determine the distance which had to be allowed around airplane model surfaces in wind tunnels so that obstruction to the wind would not affect velocity over the surface.

The A. S. H. V. E. Laboratory undertook to make a study of wind velocity gradients near a wall sufficiently large to approach somewhat the dimensions of the side of a building. Difficulty in finding winds of the desired velocities over

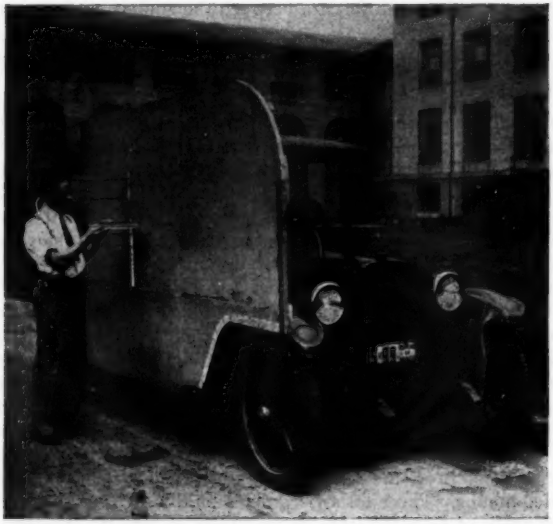

FIG. 3. TEST WALL WITH PITOT TUBES IN PLACE FOR STUDYING WIND VELOCITY GRADIENTS

a reasonable period of time made it impractical to depend on natural wind. It was also deemed impractical to produce an artificial wind of the velocities desired over a comparatively large wall and for a considerable distance away from it so as to get a natural gradient effect.

It was early agreed by those interested in the study that the equivalent of a wind against a surface could most easily be had by moving the wall forward at the desired speed while attached to the side of a car. However, there was considerable doubt concerning the possibility of measuring the small pressures set up at the various distances from the wall with gages located on the moving machine.

Preliminary tests showed that the liquid in the pressure gages moved around considerably due to acceleration and change in the level of the instrument. It was quite evident that it would be impossible to establish the desired gradient

[4] *Effect of Walls in Closed Type Wind Tunnels*, by George J. Higgins (Report 275, National Advisory Committee for Aeronautics).

by taking successive readings at different distances from the wall surface. It did seem possible, however, to make the surveys with several Pitot tubes each connected to its own pressure gage column, located on the same instrument base so that all tubes would be subjected at any instant to the same disturbing forces. It also seemed possible that these various manometer tubes and the levels of the colored oil in them might be photographed at frequent intervals for later analysis. This method of attack was attempted and proved surprisingly successful.

Fig. 3 is a photograph of the wall in place on the truck with the 10 Pitot tubes. Fig. 4 is a drawing of the wall which was 10 ft in length and 6 ft high. When it was attached to the truck, the front end was placed slightly ahead of the most forward part of the truck, so as to prevent air waves set up by the moving truck from passing around the front end of the wall. The upper front corner of the wall was curved off so as to pass through the air with less wave disturbance, and the lower front was cut away to avoid interference with turning of the front wheel.

Pitot tubes were used for measuring air velocity at the various distances from the surface. They were mounted in a vertical series 2 ft in length. The upper tube was 18 in. and the lower tube was 30 in. from the top and bottom edges of the wall, respectively. This series consisting of 10 tubes was 7 ft from the front edge and 3 ft from the back edge of the wall.

Standard Pitot tubes were used to measure the velocity at points 1 in. and farther from the surface. Special tubes made from hypodermic needle tubing having an outside diameter of 0.0720 in. and an inside diameter of 0.0521 in. were used at points from 1/16 to 2 in. from the surface. These were calibrated by comparing their dynamic pressures with those of the standard tubes at the same distance from the surface and with the same air velocity. They were single bore tubes for indicating dynamic pressures only.

A special static pressure chamber, Fig. 5, was used with the small tubes. It consisted of a brass plate ⅛ x 3 x 6 in. on the back of which was soldered a brass box 4 in. long and having a cross section 1 in. square. Two rows of seven 0.040-in. holes were drilled through the plate into the box to serve as static openings. Three ¼-in brass tubes were joined to the back of the box for manometer connections. The chamber was mounted near the small tubes so that the surface of the brass plate lay in the plane of the wall. The static pressure produced in the chamber was found to be the same as that produced by the static openings of the standard tubes for any wind velocity.

The static and dynamic connections of each Pitot tube were connected to the two legs of its inclined U-tube in the compound pressure gage. The single small dynamic tubes were each connected to one leg of one of the U-tubes. The other leg was connected to the special static chamber.

The range of distances from the surface to the dynamic orifices of the tubes used was with the small tubes at 1/16, ¼, ½, 1, and 2 in. and the large tubes at 1, 2, 6, 12, and 24 in. from the surface. The number of tubes was limited to ten by the number of manometer tubes which could be accommodated on the inclined gage platform. This arrangement gave eight different distances from the surface and by having both a large and small tube at 1 and 2 in. a check was provided on the readings of the two types of tubes. The data submitted were taken with such an arrangement.

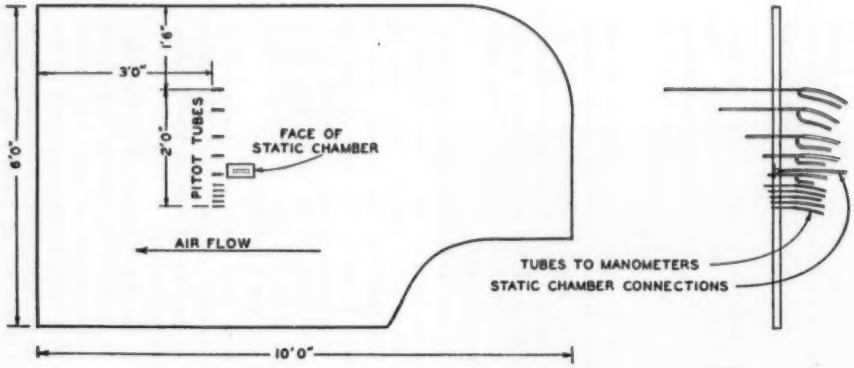

FIG. 4. TEST WALL SHOWING LOCATION OF PITOT TUBES

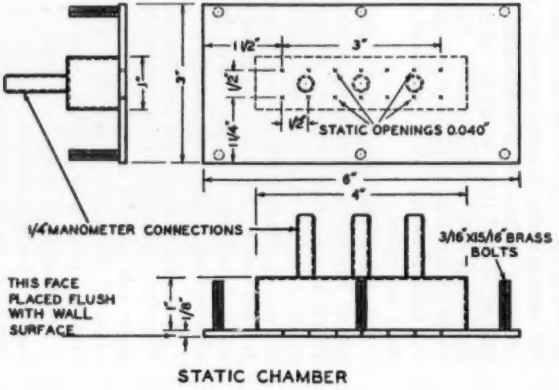

FIG. 5. STATIC CHAMBER FOR MEASURING STATIC PRESSURE AT THE WALL SURFACE

The compound pressure gage with the inclined U-tubes and the camera and lighting arrangement for photographing the tubes are shown in Fig. 6. Records of manometer readings were made with a moving picture camera using a standard 35 mm film and adjustable to make exposures at rates of from 1 to 16 per second or still exposures of any desired time.

Besides the 10 U-tubes on the inclined plane of the gage, there were also two leveling tubes, one transverse to the direction of the U-tubes which indicated any change in transverse level of the instrument or any transverse acceleration tending to change the level of the liquid in the tubes. The other leveling tube was placed parallel to and on the left hand side of the U-tubes and was so arranged that it would indicate any change in longitudinal level or longi-

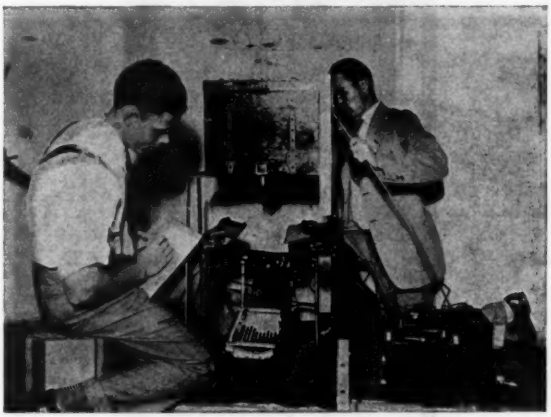

FIG. 6. COMPOUND PRESSURE GAGE AND MOVING PICTURE CAMERA FOR PHOTOGRAPHING OBSERVATIONS

tudinal acceleration tending to change the level of the liquid in the tubes. The longitudinal level was so calibrated that if a given photograph showed a deflection for this level the U-tube readings could be corrected for it. No data were used from photographs in which the transverse level showed any appreciable disturbance.

Any individual photograph taken during a test recorded the liquid levels at a particular instant when the levels might be either high or low because of the disturbing forces or lack of level. These variations in level were comparatively small and acted alternately in opposite directions. From a study of a number of consecutive pictures it was found that their average gave a consistent value.

Fig. 7 is a reproduction of one of the moving picture films showing the displacement of the red oil in the inclined U-tubes for a fairly high wind velocity. The single tube at the extreme left indicates longitudinal disturbances. The tube which indicates transverse disturbance is barely visible at the bottom.

Tests were made on a straight, level, and smooth concrete highway recently built a few miles out of Pittsburgh. During the tests the car was run as

uniformly as possible at the desired speed, allowing the levels in the various U-tubes to become fairly constant, after which a number of exposures were made. After the negatives were developed they were placed in a small projecting apparatus and thrown on a screen so that the size of the projected picture was considerably larger than the instrument board from which the picture was taken. The enlarged manometer displacements were read and corrected for the ratio between the size of the projected picture and the instrument from which the picture was taken.

Fig. 8 gives the velocity gradient away from the wall as determined for a relative air velocity of 30.7 mph (2,700 fpm). The average of many test

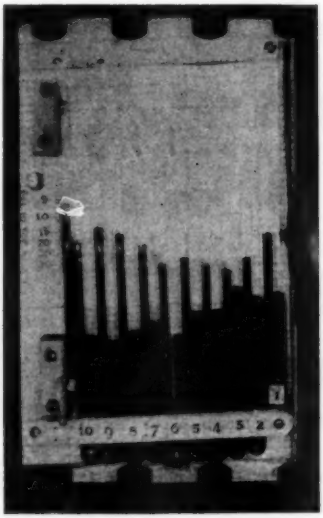

FIG. 7. MOVING PICTURE PHOTOGRAPH OF PRESSURE GAGE DISPLACEMENTS

points observed for this speed at each distance from the wall are indicated by the symbol 'Δ' where the standard Pitot tubes were used and by '$\bullet$' where the small special tubes were used. It will be noticed that one tube of each type was used at the 1 and 2-in. distances.

Similar curves were plotted for each of the several velocities studied. Points taken from all of the experimental curves similar to Fig. 8 were then plotted for each of the various distances from the wall in Fig. 9. In these curves the maximum velocity shown on the curves (similar to Fig. 8) was plotted on the X axis and the velocities measured at the respective distances X from the wall were plotted on the Y axis. Experimental points are shown for the curve $X = 0.5$. It will be noted that these points all fall remarkably close to the curve, indicating as would be expected that the series of curves of which

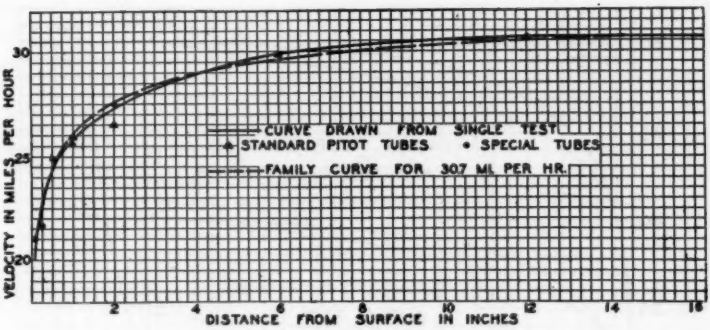

FIG. 8. WIND VELOCITY GRADIENT FOR SINGLE TEST AND FAMILY CURVE FOR THIS VELOCITY

Fig. 8 is a sample form a family of similar curves. This family of curves is plotted for different velocities in Fig. 10. The solid line curves are plotted from the curves for the various distances in Fig. 9.

Wind velocity gradients away from the test surface in the heat transfer apparatus, Fig. 1, were measured by the survey Pitot tube for both the 12-in. and 6-in. ducts. Fig. 11 gives a sample survey in the 12-in. duct at 8.75 mph and also a sample survey in the 6-in. duct at 14.75 mph. A large number of

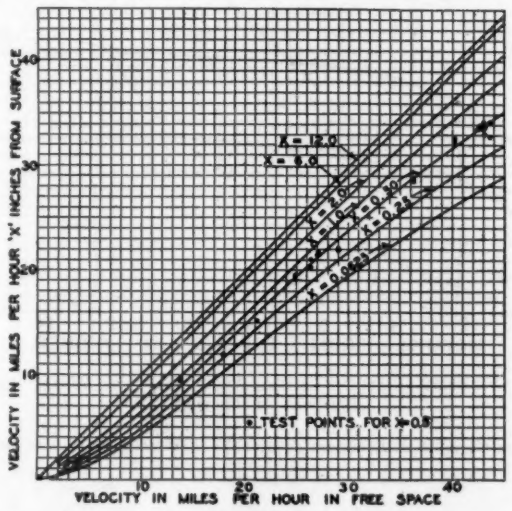

FIG. 9. RELATION BETWEEN VELOCITY IN FREE SPACE AND VELOCITIES AT POINTS DISTANT *X* FROM THE SURFACE FOR FREE AIR

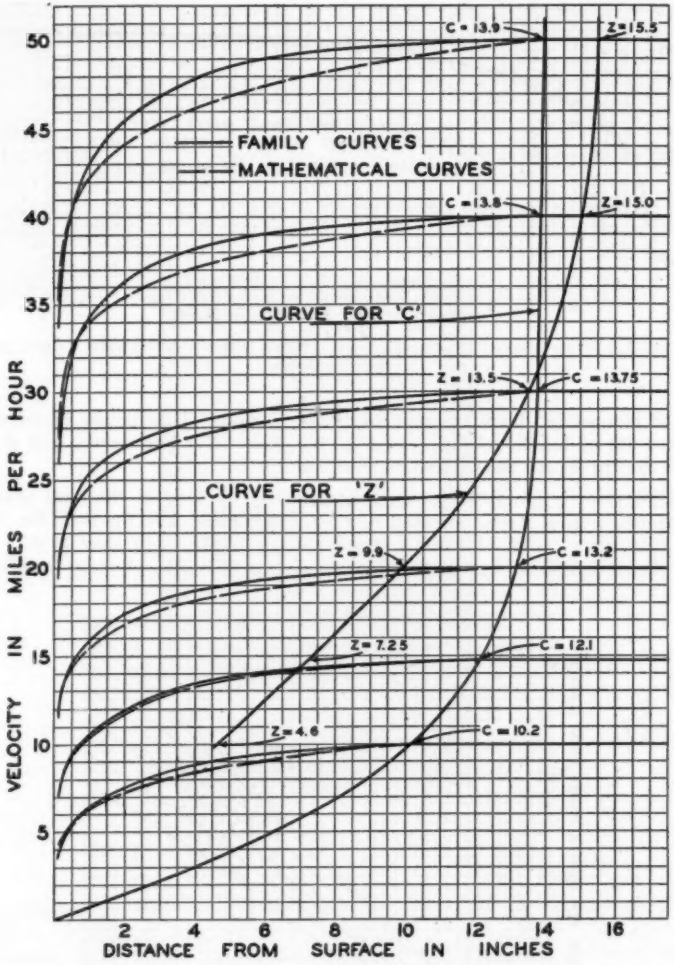

FIG. 10. VELOCITY GRADIENT CURVES AWAY FROM THE WALL. SOLID LINE CURVES FROM TEST DATA [a]

[a] Broken Line Curves from Equation $\frac{V_x}{V_c} = \left(\frac{X}{C}\right)^{\frac{1}{Z}}$ Where C and Z have the values shown.

such surveys were made and plotted. The velocity indicated by these curves at various distances X inches from the surface were plotted against the velocity in the center of the 12-in. duct in Fig. 12 and against the velocity in the center

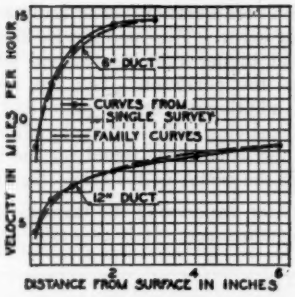

FIG. 11. VELOCITY GRADIENT CURVES IN 12-IN. AND 6-IN. DUCTS SHOWING CURVES FROM SINGLE TEST AND FROM THE FAMILY OF CURVES

of the 6-in. duct in Fig. 13. From these curves the families of gradient curves for various wind velocities in the two ducts were plotted in Fig. 14. Gradient curves in free space are also shown for comparison.

DISCUSSON OF TEST RESULTS

The curves giving the relation between film conductance and wind velocity in Fig. 2 shows a higher conductance coefficient for the rough surface at high wind velocities and apparently a slightly lower coefficient for the same surface for still air and low wind velocities. The low coefficient for the sanded surface at low velocities and still air, while indicated by the data, is so small that it

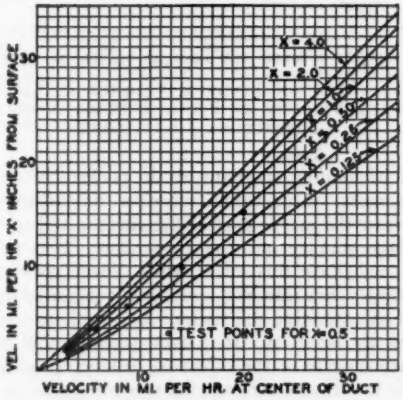

FIG. 12. RELATION BETWEEN VELOCITY AT CENTER OF DUCT AND VELOCITY AT POINTS DISTANT *X* FROM THE SURFACE FOR A 12-IN. DUCT

should not be given much consideration. The indicated difference is less than the possible experimental error. The increase in film conductance coefficients for the sanded surface at higher velocities is real and cannot be due to any possible error or unusual test condition. The few tests made on this surface after the application of additional coats of paint had smoothed out the sharp projections of the sand particles indicate that the elimination of the roughness reduced the heat loss to the same value found for the smooth pine surface with the same wind velocity in the center of the same size duct.

There is much speculation concerning the theoretical effect which roughness of surface has on film conductance. It is generally assumed that a rough sur-

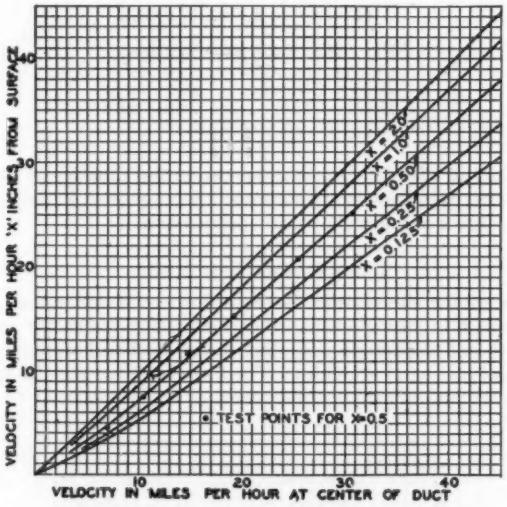

FIG. 13. RELATION BETWEEN VELOCITY AT CENTER OF DUCT AND VELOCITY AT POINTS DISTANT X FROM THE SURFACE FOR A 6-IN. DUCT

face has the effect of holding stagnant air in pockets between the high points producing the roughness. This would increase the thickness of the relatively stationary air film near the major part of the surface and increase the resistance to heat flow from the surface into the air stream. It is probable that certain types of roughness will have this effect while other types of roughness resulting from comparatively few distinct and clear cut projections on the surface will have the effect of extending the surface area without appreciably producing pockets of stagnant air between. It is even possible that these extended particles on the surface may have the effect of producing eddy currents over the surface and into the low spaces between the projections so as actually to produce greater agitation and motion of the air nearest to the surface of the wall.

Film conductance as defined and evaluated includes the heat dissipated from a surface to air by conduction and convection and by radiation to surrounding

surfaces and space. Hence the emissivity of the test surface, its absolute temperature and the emissivity and absolute temperature of the surfaces or space which can be seen by the test surface all affect the film conductance.

In determining the film conductance coefficient it is therefore necessary to take radiation into account. The coefficients here reported were determined with the parts of the apparatus in view of the test surface at an air temperature within plus or minus one degree. This was accomplished by using a low air temperature of 70 F which was the same as the temperature of the room in which the apparatus was located.

It is of interest to note the effect which variation in temperature of the surfaces in view of the test surface has on the coefficients determined. Heat dissipation from a surface is given by the formula,

$$f = \frac{H}{t_s - t_a} = \frac{W_r + W_{cv}}{t_s - t_a} \tag{2}$$

$$= \frac{0.178 \times 10^{-8} E\left(T_s^4 - T_s'^4\right) + W_c \sqrt{\frac{v + 68.9}{68.9}}}{t_s - t_a} \tag{3}$$

where

H = the heat transfer from the surface in Btu per square foot per hour for the actual temperature difference.
t_s = the temperature of the test surface.
t_a = the temperature of the air unaffected by the surface.
T_s = the temperature of the test surface in degrees absolute.
T_s' = the temperature of the surfaces in view of the test surface in degrees absolute.
W_r = heat dissipated from the surface by radiation in Btu per square foot per hour.
W_c = heat dissipated from the surface in still air by convection in Btu per square foot per hour.
W_{cv} = heat dissipated from the surface by convection for the wind velocity v in Btu per square foot per hour.
E = the emissivity of the surface.

Equation (3) may be written, $f =$

$$\frac{0.178 \times 10^{-8} E\left(T_s^4 - T_s'^4\right)}{t_s - t_a} + \frac{W_c \sqrt{\frac{v + 68.9}{68.9}}}{t_s - t_a} \tag{4}$$

The first term of this equation expresses the rate of heat dissipation by radiation. For the condition of the tests here reported the test surface was maintained at 90 F and the duct wall surfaces in view of the test surface were maintained at 70 F, plus or minus 1 deg. Assuming a value of 1 for E, the value of the radiation term in equation (4) becomes

$$\frac{0.178 \times 10^{-8} (90^4 - 70^4)}{90 - 70} = \frac{22.41}{20} = 1.1205$$

If the temperature of the surrounding wall surfaces fell by 10 deg without any change in the air temperature the value of the term would become,

$$\frac{0.178 \times 10^{-8} (90^4 - 60^4)}{90 - 70} = \frac{32.75}{20} = 1.6337$$

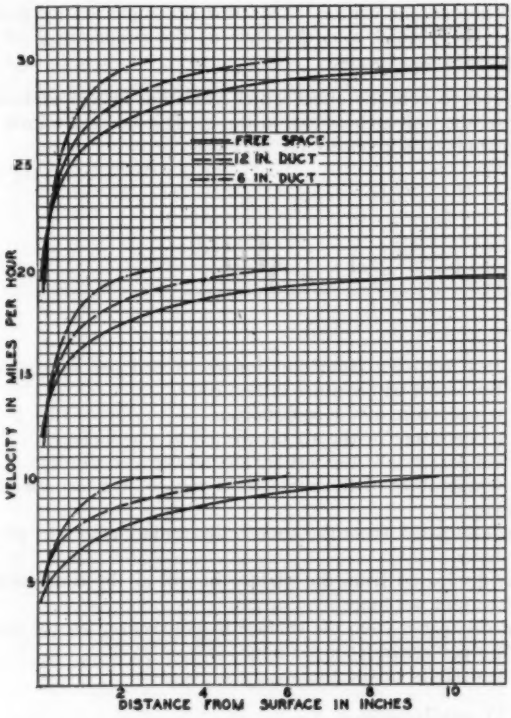

FIG. 14. VELOCITY GRADIENTS FOR A SURFACE IN FREE SPACE IN A 12-IN. DUCT AND IN A 6-IN. DUCT FOR WIND VELOCITIES OF 10, 20 AND 30 MPH

A small error in the temperature of the surrounding surfaces would, therefore, make an insignificant error in the value of f.

Curves in Fig. 2 for the smooth pine surface in the 6-in. and the 12-in. ducts indicate a higher rate of heat transfer for the same wind velocity in the center of the smaller duct. This may result from a steeper temperature gradient and therefore a lower air temperature near the true surface, or it may result from a steeper wind velocity gradient near the true surface. It is probable that both of these factors operate simultaneously to increase the rate of heat transfer

from the surface in smaller ducts for the same wind velocity at the center. It is axiomatic that a steeper wind velocity gradient near the surface will automatically result in a steeper temperature gradient. Temperature gradient curves for wind velocities of 1, 5, and 34 mph in the 12-in. duct are plotted in Fig. 15. These curves show steeper temperature gradients near the surface in the same duct for the higher wind velocities. Comparison of the curves for a 34-mph wind in the two ducts shows that the same wind velocity in the smaller duct is accompanied by a steeper temperature gradient. Wind velocity

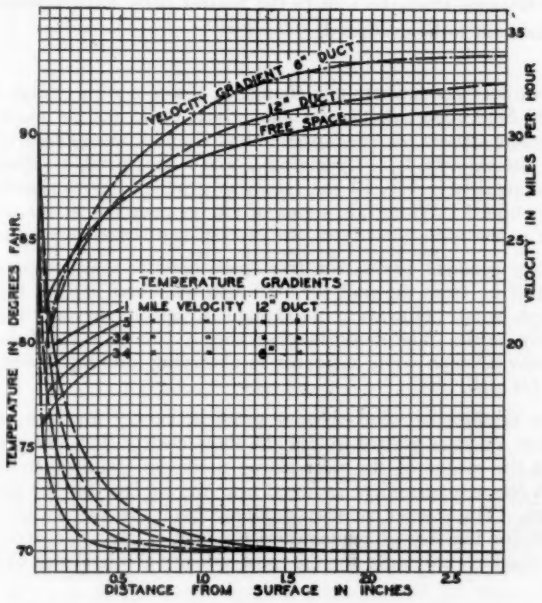

FIG. 15. TEMPERATURE GRADIENTS FOR WIND VELOCITIES OF 1, 5 AND 34 MPH IN A 12-IN. DUCT AND 34 MPH IN A 6-IN. DUCT. VELOCITY GRADIENTS FOR A 34-MPH WIND IN THE TWO DUCTS AND IN FREE SPACE

gradients near the surface are also shown for a wind velocity of 34 mph in the center of the 6-in. and 12-in. ducts and in free space.

It is of particular interest to note how near the warm surface the temperature of the air becomes constant and unaffected by the surface. Except for velocities under 5 mph the temperature gradient does not extend appreciably beyond 1 in. and for velocities as low as 1 mph the gradient does not extend beyond 1½ in. from the surface. These curves indicate that for these conditions an air temperature determined 1½ in. or more away from the surface is sufficiently accurate and unaffected by the surface for use in calculating heat transfer coefficients.

The National Advisory Committee for Aeronautics developed the following equation for velocity gradient curves resulting from their studies:

$$\frac{V_x}{V_e} = \left(\frac{X}{C}\right)^{\frac{1}{z}} \tag{5}$$

where

V_e = the velocity at sufficient distance from the surface to be unaffected by it.

V_x = the velocity at a point X distance from the wall surface.

C = the distance from the wall to the nearest point having velocity V_e.

z = a constant to be evaluated.

Their work does not indicate any change in C and z with change in velocity. In the study here reported, the formula given could not be made to fit the velocity gradient curves for free space unless C and z were given values which varied with the velocity V_e. By giving C and z the variable values shown by the curves so marked in Fig. 10, the family of gradient curves shown by broken lines were obtained. While this series of curves does not fit the experimental data as well as the solid line curves, the discrepancy is not very great.

The velocity gradient curves for free space indicate that a constant velocity is reached 10.2 in. from the wall for a 10-mph wind and 13.9 in. from the wall for a 50 mph wind. However, the change in velocity beyond 6 in. becomes comparatively small. The error in observing a 10-mph wind 6 in. from the surface is only 0.5 mph while the error in observing a 50-mph wind 6 in. from the wall is 1.0 mph.

The three families of wind velocity gradient curves for free space, 12-in. duct, and 6-in. duct, as plotted in Fig. 14 show clearly that any given velocity measured in the center of the duct shows a steeper gradient for the smaller duct while a less steep gradient is found near the surface for the same velocity in free space. The steeper velocity gradients near the surface of the smaller duct account for the steeper temperature gradient. They also account for the higher film conductance curve for the same velocity at the center of the smaller duct.

It is probable that an equation expressing accurately the true velocity gradient curve near a surface, together with the air temperature in the free unobstructed space, and the temperature of the surface, would prove sufficient to determine accurately the rate of heat transfer. The methods available for making the necessary measurement of wind velocities for developing the wind velocity gradient curves and their equations are, however, not sufficiently accurate for this purpose, largely due to inability to properly evaluate the true effective distance between the Pitot tube dynamic orifice and the surface and also due to the disturbing effect which the close proximity of the Pitot tube and the wall have on the wind velocity at this point.

A study of the wind velocity gradients for the 6-in. and 12-in. ducts in Fig. 14 in comparison with the film conductance curves for the smooth pine surface in the 6-in. and 12-in. ducts, Fig. 2, will give some indication of the relation between the distance from the surface at which the wind velocity is measured and the film conductance coefficient obtained.

Wind velocities of 10, 20, and 30 mph as measured in the center of the 12-in. duct give velocities of 9.1, 19.1, and 28.9 mph respectively measured at 3 in. from the surface. The ratios of the velocities at the center of the 6-in. duct to the respective velocities measured at the same distance, or 3 in., from the surface in the 12-in. duct are as follows:

$$\frac{10}{9.1} = 1.099$$

$$\frac{20}{19.1} = 1.044$$

$$\frac{30}{28.9} = 1.039$$

For these same wind velocities the ratios of the film conductance coefficients measured in the 6-in. duct to the respective film conductance coefficients measured in the 12-in. duct compare closely with the velocity ratios and are as follows:

$$\text{For a 10-mph velocity } \frac{5.25}{4.81} = 1.091$$

$$\text{For a 20-mph velocity } \frac{8.50}{7.75} = 1.098$$

$$\text{For a 30-mph velocity } \frac{11.20}{10.25} = 1.092$$

If it may be assumed that the same relationship holds for other sizes of ducts and for free space, then the coefficients for any velocity as measured at the center of different ducts may be expressed as proportional to the velocity in that duct measured at 3 in. from the surface. Coefficients for a smooth pine surface and a 20 mph wind measured in a 6-in. and a 12-in. duct and in free space would vary as the respective velocities measured at 3 in. from the surface or as 20, 19.1 and 18.1. Based upon a coefficient of 8.5 determined in the 6-in. duct, the coefficients measured in a 12-in. duct and in free space should be 8.1 and 7.7 respectively. The value of the coefficient measured in the 12-in. duct was 7.75. Hence the application of the foregoing reasoning gives a fair approximation.

The curves indicating the relation between film-conductance coefficient and wind velocity developed at the laboratory and presented in Fig. 2 of this paper are shown in Fig. 16 together with similar curves reported by Rowley, Algren and Blackshaw [5] for an 80-F mean temperature. Both investigations give data for a smooth finish white pine surface. The values in this report giving curves for the 6-in. and the 12-in. ducts were determined with a 20-deg. temperature difference and an 80-F mean temperature. Both reports include data determined in a 6-in. duct with an 80-F mean temperature which should check. It is observed, however, that the Laboratory curve *C* for an 80-F mean tempera-

[5] *Surface Conductances as Affected by Air Velocity, Temperature and Character of Surface*, by F. B. Rowley, A. B. Algren, and J. L. Blackshaw (A. S. H. V. E. TRANSACTIONS, Vol. 36, 1930).

ture determined in a 12-in. duct coincides with Rowley's curves *G* for an 80-F mean temperature determined in a 6-in. duct. This discrepancy is somewhat

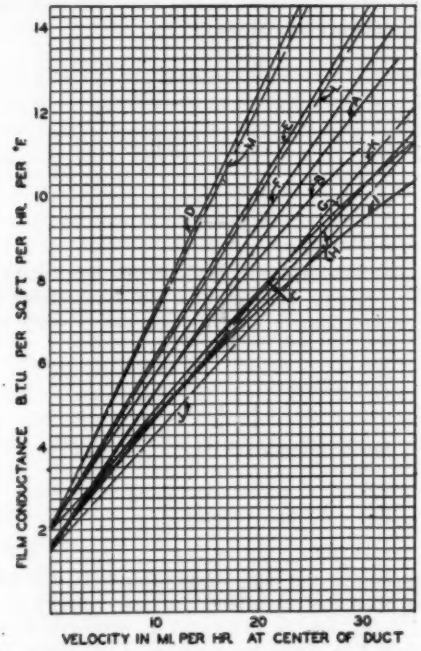

FIG. 16. FILM CONDUCTANCE COEFFICIENTS FOR STILL AND MOVING AIR AS DETERMINED IN THIS STUDY AND REPORTED BY ROWLEY AND OTHERS [a]

[a] Curve *A* Sanded Surface ..... This Report
B Smooth Pine (6-in. duct) ..... This Report
C Smooth Pine (12-in. duct) ..... This Report
D Stucco ..... Rowley, Algren and Blackshaw
E Brick and Rough Plaster ..... Rowley, Algren and Blackshaw
F Cast Concrete ..... Rowley, Algren and Blackshaw
G Smooth Pine ..... Rowley, Algren and Blackshaw
H Smooth Plaster ..... Rowley, Algren and Blackshaw
I Glass and White Gloss Paint .. Rowley, Algren and Blackshaw
J $f = 1.4 + 0.28V$
K $f = 1.6 + 0.3V$
L $f = 2.0 + 0.4V$
M $f = 2.1 + 0.5V$

larger than the possible experimental error but may be accounted for by the difference in pine surfaces studied.

The entire group of curves show a marked similarity. All the materials included in the two studies show still-air film-conductance coefficients ranging from 1.4 to 2.1. All the curves plotted against wind velocity show a similar

tendency. The smoother surfaces show a lower coefficient for the same wind velocity and a slight curvature.

For wind velocities up to 35 mph, a straight line would approximate any of the curves to a fair degree of accuracy. The curve J approximating the lower limit of the entire series gives a relationship between film conductance and wind velocity satisfied by the equation $f = 1.4 + 0.281v$, where v is the wind velocity. The curve D for stucco is satisfied by the equation $f = 2.1 + 0.515v$. For a rough approximation, the curves K, L, and M having the following simple equations could be accepted as representing the following general types of surfaces: For curve K, representing smooth wood and plaster, $f = 1.6 + 0.3v$. For curve L representing cast concrete surfaces and fairly smooth brick surfaces, $f = 2 + 0.4v$. For curve M, representing rough stucco and other similarly rough surfaces, $f = 2.1 + 0.5v$.

The several curves for the different surfaces studied in the two investigations indicate rather clearly that roughness of surface is the main factor affecting film conductance coefficients for any type of material used in building construction. A further study and gradation of surfaces with respect to roughness should therefore make possible the acceptance of a film conductance coefficient for new types of surfaces without the necessity of actually testing them. In the meantime, the rough approximation of the surfaces for which curves and formulae are given may be of some practical value.

SUMMARY AND CONCLUSIONS

1. This report presents wind velocity gradient curves away from a wall in free space and away from the surface of rectangular ducts which are 6 in. and 12 in. in width. Film conductance coefficients for still and moving air for a sanded surface in a 12-in. duct and for a smooth pine surface in both a 6-in. and a 12-in. duct are also presented.

2. Wind velocity gradients for free space extend to distances of 10.2, 13.2, and 13.7 in. from the wall for velocities of 10, 20, and 30 mph, respectively.

3. For the same velocity in the center of rectangular ducts of different sizes the gradient is steepest near the surface of the smaller duct. This is accompanied by a steeper temperature gradient and a higher film conductance.

4. Temperature gradients are shown to vary with the size of the duct and the wind velocity. For a 20-deg temperature difference and velocities of 5 mph and higher, the temperature gradient lies within 1 in. of the surfaces. For velocities down to 1 mph, the gradients extend to as far as 1½ in. from the surface.

5. Film conductances for a number of surfaces reported here and elsewhere are shown to vary between 1.4 and 2.1 for still air and between the limits of the simple equations:

$$f = 1.4 + 0.281v$$

and

$$f = 2.1 + 0.515v$$

where v = the velocity in miles per hour.

Film conductance is largely a function of roughness for the surfaces considered and can probably be expressed from a gradation of roughness of surfaces. A

rough approximation of such a gradation and the equation for the coefficients are presented.

DISCUSSION

PRESIDENT HARDING: About 15 years ago at the University of Illinois several engineers decided it would be a fine thing to determine the effect of wind velocity on the surface coefficient. Apparently at that time data were not available, as far as building material was concerned, and accordingly they set up test apparatus. The wind velocity coefficients which were determined have been in use ever since. Having had a part in these tests, it occurred to me a few years ago that it might be wise to check some of those results as it was evident that little correlation existed between the wind velocities reported by the Weather Bureau, and the actual wind velocity over the surface of structures.

The Weather Bureau reports, for example, a 25-mile wind velocity on the top of a 10-story building. Just what relation does the wind velocity report of the Weather Bureau bear to actual wind velocities over a building wall surface? While serving as chairman of the Committee on Research I persuaded the Director to put this subject on the program. It is evident, of course, that considerable more work will be necessary on this particular project before the correlation between the wind velocities as reported by the Weather Bureau and actual wind velocities over various surfaces is satisfactorily determined.

PROF. A. C. WILLARD: The problem of determining surface coefficients, where we have convection and conduction so intimately intermingled, is a most difficult one. The attempt here has been to get something that would have a practical application. The data referred to by President Harding, I believe, substantially check at the same velocities that the authors of the paper have secured in these later and more extended tests at the Research Laboratory of the Society.

The subject is by no means of limited interest. It is of importance not only in building construction, but in the loss of heat from ducts of small sizes as in buildings and also of very large size as in tunnels and possibly in mines and in subway work. The establishment of more fundamental data are, of course, desirable and it may be possible that we can, considering the case of the question of the value of insulating surfaces using air cell construction, make the investigation more fundamental in its character, or at least determine some of its fundamentals. When that is done, the subject will be of general interest to the engineering professions dealing with air flow, and air motion as well as heating and ventilation.

There is a chance here for us to cooperate with the work that is being done in the field of aeronautics. No doubt many valuable data have been secured which we could apply and which would save a lot of work by our Laboratory. The function of the Laboratory should be to correlate whatever data are already available, with the information we may have before final conclusions are drawn.

President Harding is to be congratulated on the skill with which he anticipated just what might happen when we put an enclosure around a building column and tested it for the effect of air velocity on convection and conduction temperature gradients of the wall, and that 15 years later we find the Society's Research Laboratory in general substantiates the results secured in this simple apparatus. President Harding has always been successful in using the minimum amount of equipment to get a maximum amount of data, which are of immediate practical use to the engineering profession.

No. 896

OFF-PEAK SYSTEM OF ELECTRIC HEATING FOR BUILDINGS

By ELLIOTT HARRINGTON,[1] SCHENECTADY, N. Y.
MEMBER

MANY years ago, Dr. Charles P. Steinmetz predicted that some day homes would be built without chimneys and heated entirely by electricity. Today, the average informed man knows that he can heat his home electrically if he chooses to do so, but he immediately associates exorbitant cost with this operation. He knows that electric heat would be clean, safe, convenient, silent, and reliable. But his knowledge usually ends with the 500-watt radiant heater which he sets in his bathroom on a cold morning. He has learned that when he uses this radiant heater consistently he receives a materially increased electric power bill at the end of the month and he has only electrically heated one small room a part of the time. However, the prophecy of Dr. Steinmetz is much nearer an economic and practical realization today than is generally realized, and off-peak power appears to be the solution of the problem.

During most of the daytime and evening when the electrical consumers are drawing power at or near the full capacity of the power station, any increase in load immediately requires an increased investment in generating equipment. However, during certain hours of the twenty-four, the demand on the generating equipment is very low. The particular hours will vary, of course, for various localities, depending upon the characteristic use of energy in that particular territory. These hours of small usage, or *off-peak* hours, as they are known, in practically all territories include the hours from 10:30 P. M. to 7:00 or 8:00 A. M., and in most territories include certain periods during the day. These hours of usage are very firmly established by the type and habits of the electrical customers, and it is possible to adapt the use of apparatus to the *off-peak* hours characteristic of any particular locality. The power generated during these off-peak hours involves costs for fuel, incidental labor, and supplies, but does not involve additional investment cost for generating equipment.

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Presented at the 37th Annual Meeting of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, Pittsburgh, Pa., January, 1931.

TABLE 1. ENERGY USED FOR ELECTRICALLY HEATING EIGHTEEN HOUSES IN FORT WAYNE, IND.
Oct. 15, 1929, to May 15, 1930

House	Description of House	New or Old House	Insulation	Cubical Content Heated	Kilowatts Installed	Net Kilowatt-Hours Used [a]
1	Brick veneer	Old	Walls and ceiling	44,840	80	122,310
2	Brick veneer	Old	Ceiling and double glass	32,499	60	85,970
3	Brick veneer	New	Walls, ceiling and double glass	28,255	50	35,695
4	Clapped frame	Old	Walls, ceiling and double glass	28,124	50	78,750
5	Stucco frame	Old	Walls, ceiling and double glass	26,250	60	70,615
6	Brick veneer	New	Walls and ceiling	22,650	40	35,833
7	Brick veneer	New	Walls and ceiling	22,400	60	51,443
8	Brick veneer	New	Walls and ceiling	21,920	60	55,146
9[b]	Stone	Old	Ceiling and double glass	21,277	60	68,817
10	Clapped frame	Old	None	20,020	50	60,690
11	Clapped frame	Old	None	20,020	50	54,922
12	Clapped frame	Old	Walls and ceiling	15,700	40	37,800
13	Clapped frame	Old	Walls and ceiling	15,620	50	40,025
14	Brick veneer	New	Walls, ceiling and double glass	14,000	30	25,222
15	Shingle frame	New	Walls, ceiling and double glass	13,690	30	13,900
16	Brick veneer	New	Walls, ceiling and double glass	13,170	30	19,080
17	Clapped frame	New	Walls, ceiling and double glass	12,420	30	14,990
18	Shingle frame	New	Walls, ceiling and double glass	9,130	30	17,170

[a] Some variation between size of house and kilowatt-hour consumption may be accounted for by house temperatures maintained by individual householders, and other unusual operating conditions.
[b] House No. 9 located in Philadelphia, Pa.; all others in Ft. Wayne, Ind.

Hence, as the telegraph company is ready to send a day letter or a night letter at a reduced rate when the wires are not busy, so the central station is ready to supply electric energy at a low rate when the apparatus which they already have installed is relatively idle. The problem of electric heating of buildings, therefore, becomes one of storing this electric energy purchased during the low rate period and using this stored energy as required.

A number of different methods have been tried for storing this electrical energy and using it later in the heating of buildings. The original and most obvious method was an effort to store and use the electricity itself by charging

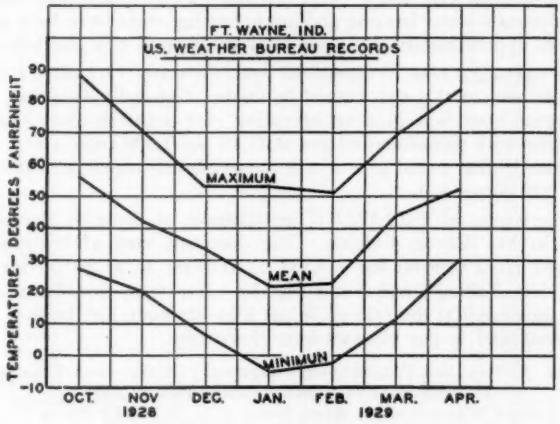

FIG. 1. OFFICIAL OUTSIDE TEMPERATURE RECORDS FOR SEASON OF HOUSE HEATING TESTS

storage batteries, but the first cost of the apparatus for any reasonable storage capacity was prohibitive.

Then media were tried for storing this energy in the form of electric power converted into heat. Sand, crushed rock, cast iron, soapstone and water have all been used with varying degrees of success. Probably water is the most acceptable because it is universally available, costs nothing and, since it is a fluid, it can be circulated. It is immediately adaptable to the conventional hot water radiation system.

COST

Before discussing the apparatus in detail and its application to various heating problems, consider the question of paramount importance, namely the cost of operating.

Of course, the first cost of the installation varies with the characteristics of the building. However, the installation cost of a typical residence of 22,000 to 28,000 cu ft is about comparable to the cost of a high grade boiler, oil burner, control, and a 1,000-gal outside oil tank. A complete gas heating system, consisting of gas boiler, burner, and control, would cost about two-thirds as much as the electric system, and a complete coal burning system about one-

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third as much. In a new building, the first cost of the radiation can be reduced somewhat with electric heating because the higher temperature of the circulating water requires less standing radiation, and with forced circulation the distributing pipes can be smaller.

The operating cost depends upon a number of variables, among which are the power rate obtained from the central station and the characteristics of the building to be heated. Central stations are rapidly accepting electric heating as a thoroughly workable means of increasing their load factor (ratio of average load to maximum load demand). In many states—notably Florida, Georgia, Illinois, Indiana, Iowa, Ohio, Virginia, Pennsylvania and New Jersey —*off-peak* domestic water heating and house heating rates have been established ranging from approximately 0.75c per kilowatt hour to 1.5c per kilowatt hour.

Since the operating cost is dependent upon so many variables, a broad and accurate statement of this cost cannot be made. Generally speaking, a rate of 1c per kilowatt hour will give an operating cost approximately equal to the cost of heating with manufactured gas at $1.10 per 1,000 cubic feet. A rate of ½c per kilowatt hour would give a cost approximately equal to that of anthracite coal at $18.00 per ton.

During the winter of 1928-1929 17 installations of domestic electric heating were made in Ft. Wayne, Indiana. The operating cost of heating these 17 homes ranged from $190.00 for a 9,130 cu ft home to $1,300.00 for a 44,840 cu ft home (see Tables 1 and 2 and Fig. 1). For these installations, off-peak power was purchased at the rate of 2c per kilowatt hour for the first 400 kwhr used per month and 1c per kilowatt hour thereafter.

TABLE 2. AVERAGE TEMPERATURES DURING JANUARY AND FEBRUARY IN FORT WAYNE, INDIANA, FROM 1925 TO 1929

Based on records of Ft. Wayne branch of U. S. Weather Bureau

Year	Temperature (Degrees F) January	February
1925	23.6	34.4
1926	27.1	30.4
1927	25.2	35.2
1928	26.8	29.7
1929	21.4	23.1

In considering the operating cost of electric heating, it must be remembered that in this system electricity is converted into heat at an efficiency of 100 per cent. Every kilowatt-hour provides 3,415 Btu of heat energy and every one of these Btu is released into the home. Therefore, the electric meter immediately becomes a Btu meter. Furthermore, it is not necessary to draw any air from out of doors, heat it to room temperature, use it to support combustion, and then dispose of it through the chimney.

ADVANTAGES

The advantages of heating electrically are generally known, but it is well to review them briefly as follows:

1. *Safety.* As safe as the electric wiring system now installed in 19,000,000 homes, and it operates without flame or combustion of any kind. No combustible is brought into or stored in the house.

2. *Reliability.* There has never been an electric power famine, and prolonged power failure in a given locality is almost unknown.

3. *Cleanliness.* No solid, liquid or gaseous fuel comes into the building, and no ashes go out. No smoke, fumes or soot are released in or around the building.

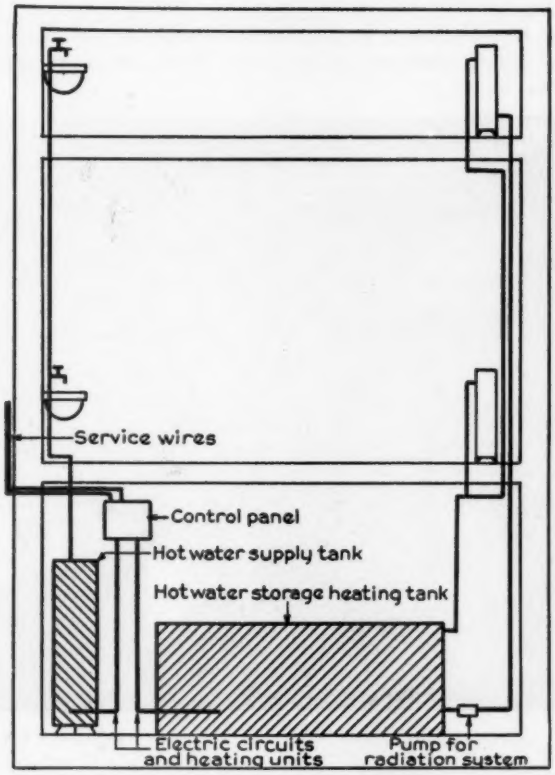

FIG. 2. DIAGRAM SHOWING THE ESSENTIAL FEATURES AND TYPICAL ARRANGEMENT OF UNITS OF AN "OFF PEAK" ELECTRICAL HEATING SYSTEM

4. *Quietness.* The only moving mechanism is the heat circulating device which can be made to operate without noise.

5. *Availability.* The electric heating unit, because it does not involve open flame, and because the storage tank can be made of different shapes—that is, tall and vertical or broad and horizontal—can be placed in almost any desired part of a dwelling or other building.

6. *Low Maintenance.* Absence of combustion tends to reduce maintenance to a minimum.

DESCRIPTION

Briefly, the apparatus for the electric heating of buildings using a hot water system, consists of the following (see Figs. 2, 3, 4 and 5):

1. Heat insulated tank for the storage of hot water.
2. Electric immersion units for heating the water in the tank.
3. Control panel containing main switch, time and temperature control switches.

FIG. 3. A COMPLETE HOT WATER UNIT INSTALLED IN A SPACE 6 FT. BY 10 FT.

4. Circulating pump (for warm air systems, a fan circulating air around the heat storage tank and through the hot air ducts would be substituted for the circulation pump).

OPERATION

The operation of the system is as follows: During the *off-peak* period an electrically driven time switch energizes the immersion units which heat the water in the storage tank to 250 F (for warm air systems the water is heated to 300 F). When the predetermined temperature is reached, a pressure switch cuts off the power with enough heat stored in the tank to heat the building in question for 24 hr under the most severe weather conditions. When the room

thermostat calls for heat, it starts the circulating pump which circulates hot water from the storage tank to the radiators. With warm air, the fan is started, thereby circulating air through a jacket around the hot water storage tank or through a unit heater to the registers. When proper room temperature is obtained, the thermostat shuts down the circulating unit. The heat storage tank is so well insulated that the heat loss from it is very low. The small amount of heat given off by the heat storage warms the basement or whatever part of the building houses the storage tank.

Of course, heat insulation of the building is an important part of any heating problem and it is particularly important with electric heat. It is a well recog-

FIG. 4. A TYPICAL WARM AIR ELECTRIC HEATING UNIT

nized fact that the installation of the proper thickness of heat insulation, together with double windows and weather strips, will reduce the heat losses from the building about 50 per cent.

DOMESTIC WATER HEATING

An *off-peak* water heater has been developed as a part of this system and can well be considered with it. This water heater (Fig. 6) is particularly important from the standpoint of the central station, since to improve the load factor during the summer months, the *off-peak* period must be utilized to maintain the

low power rate throughout the year. This *off-peak* water heater operates on the same principle as the system for building heating—that is, during the *off-peak* period, water is heated in an insulated storage tank to a temperature of about 200 F. The water is then drawn from this tank as needed during the day through a thermostatically controlled mixing valve which mixes sufficient cold water with the water from the heat storage tank to produce the desired temperature of perhaps 130 to 140 F. When this water heater is installed with

FIG. 5. A HOT WATER ELECTRIC HEATING UNIT INSTALLED IN A PIT AND WITHOUT A BASEMENT

the electric heating equipment for the building, the same time switch and electric meter controls both units.

APPLICATION

In the past three seasons approximately fifty complete installations of electric heating systems have been made in various parts of the United States and have operated successfully over the range of climatic conditions existing.

In addition to the heating of homes, a number of installations have been made in sub-stations, and commercial type public utility buildings. It is significant to note that power companies in New Jersey, Ohio, Indiana, and Virginia are heating their own office buildings and sub-stations economically with the same energy which they generate, and other power companies are now making similar installations.

This thermal storage form of electric heating possesses characteristics of flexibility and safety which will meet conditions encountered in present day buildings. As an illustration, the telephone company having small, unattended buildings largely filled with delicate automatic apparatus, which has to be maintained at a reasonably uniform temperature, must have apparatus which is safe from the standpoint of fire hazard, which is free from the products of combustion, so far as the air within the heated space is concerned, and must have an automatic form of heat which can be left for hours or days in an unattended

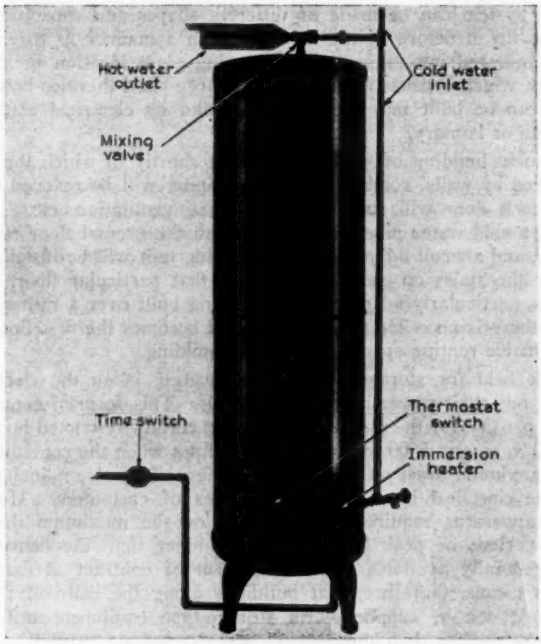

FIG. 6. OFF PEAK WATER HEATER

building. Furthermore, these heated spaces may be of a few hundred cubic feet in volume to several thousand cubic feet. The thermal storage form of electric heating meets all of these conditions. Storage tanks from those of possibly two to three hundred gallons to several thousand gallons capacity can be supplied to meet the demands of the building. The source of heat—that is, electricity—is enclosed in approved iron conduit protected with approved switches and service cut-outs to meet the requirements of the *National Board of Fire Underwriters*. The automatic controls for regulating the heat can be set to maintain the temperature desired, and there is no combustion in the building. Therefore, the entire equipment meets, in every respect, the rather

rigorous demands of this particular kind of an installation. This example illustrates the conditions which can be met.

Similarly, this form of electric heating undoubtedly will have a very definite effect on the architecture of homes in the future, because of its inherent characteristics of safety, freedom from combustion, and flexibility in its possibilities of installation. It is entirely possible to build a home without a cellar, which is now necessary, largely for housing a heating plant and storing fuel, and without a flue for taking away the products of combustion from the present types of heating plant located in the cellar. Inasmuch as the storage tanks for this heating system can be made of different shapes and capacities, they can be built into the structure of the house in such a manner as to eliminate the present exaggerated importance of the heating plant location in a house. A heating plant which consists essentially of a large-sized thermos bottle of varying shape, can be built into the first floor, and its electrical controls placed in the kitchen or laundry.

An apartment building of steel will be built shortly in which the amount of space required by walls, conduit, ducts, and piping will be reduced to a minimum, and each floor will supply its own heat, ventilation, etc. An electric conduit and a cold water pipe will be run from the ground floor to the top of the building, and a small *off-peak* electric heating unit will be installed in waste space under the stairs on each floor to heat that particular floor. This type of heating is particularly adaptable to a building built over a railroad right-of-way where there is no cellar and the basement becomes the first floor, which is the most valuable renting space in the entire building.

A possible field for storage-type electric heating is on the electrified railroads which buy their power on a *demand* basis. This form of contract carries a charge or penalty for the power taken during certain restricted hours, such as 7:00 to 9:00 A. M. and 5:00 to 7:00 P. M., at times when the commuting service makes the maximum draft on the electrical lines. This also coincides with the draft on the electrical lines from other types of customers. However, the amount of apparatus required is predicated on the maximum draft coming during the periods of peak load. At hours other than the demand periods, energy is generally available under this form of contract at extremely low rates. This means, therefore, that buildings along the railroad right-of-way requiring heat can be supplied with storage-type equipment utilizing cheap energy at hours other than the demand periods, and the energy can be taken directly from the lines running along the right-of-way.

An electrically driven ship is now under construction where the heating will be done electrically, using the Diesel electric propulsion equipment to charge the heat storage system when the boat is not under way.

CONCLUSION

Under present conditions, and considering the load curves that are characteristic of many electric properties, there is adequate *off-peak* power available to supply from two to five per cent of the homes in any territory. The continued sale of additional lighting appliances and the many other convenient motor-driven and heating appliances for the home all tend to increase the peak demand and, correspondingly, make more *off-peak* energy available. This means that from year to year, with the present steadily increasing use of all

forms of electric service, there will be made available an additional *off-peak* capacity which can be used for this form of home heating. This can be considered as a form of heating providing a maximum of safety and the utmost of convenience and cleanliness. It can be installed in place of any other form of heat and in many places where other forms of heat cannot be used.

DISCUSSION

SELBY HAAR (WRITTEN): The subject of Mr. Harrington's paper is of much interest at this time when every form of reduction of expenditures in industry is being thoroughly studied.

A comparison of the thermal storage system with a steam heating plant for a property consisting of one large building and several smaller ones not compactly arranged, showed that while the steam plant, which would contain two 400-hp boilers, is more economical in annual charges for the large building than an electric heating system, the electric equipment is more economical for the outlying buildings on account of the omission of long steam supply pipes and pipe trenches and the elimination of capital costs and fixed charges and the heavy losses of heat in the supply lines. In this comparison electrical energy was charged at considerably less than 1¢ per kilowatt-hour, and 5 per cent was allowed for interest on borrowed money.

The work of Mr. Harrington and others is marking out a definite field for thermal storage heating systems determined in some cases by economic considerations, in others by the convenience, cleanliness and other special characteristics of electric heat.

DELTON T. WABY (WRITTEN): After reviewing Mr. Harrington's paper, which is certainly a careful coverage of the title, I am inclined to believe that I can best serve your Society by adding our experiences to those already related.

The Public Service Company of Northern Illinois has, for the last 3 years, operated an off-peak, experimental electric house heating plant at the George R. Mitten residence, 39 North Drexel Ave., La Grange, Ill. This installation (a warm air unit in a house with a volume of 19,000 cu ft) was, I believe, the second of this kind to be installed in this country. The plant is a typical Hall system, using water as the heat storage medium. It was installed to give the greatest flexibility from an operating standpoint, thus making it possible to obtain data of the greatest value in determining the practicability of off-peak thermal storage electric heating of homes.

After 2 full seasons of operating and 4 months to date in the 1930-31 heating season, our experience shows the many advantages of electric heating as outlined by Mr. Harrington. The item of maintenance appears to be almost a zero quantity. When consideration is given to the elements of the system, the heating units, the control panel, the heat storage tank, and heat circulating system, which are designed for operation under conditions far more severe than those encountered in the operating cycle of an off-peak heating plant, it is evident why this item is of such little importance.

Table A gives the monthly energy consumptions for house heating for the 1928-29 and 1929-30 heating seasons.

Table B compares the first 4 months of each year with *monthly* average Weather Bureau temperature records for Chicago. A considerable reduction will be noted in the 1930 figures over both 1928 and 1929. It is possible to find corresponding months in the 2 years when average temperatures were about equal. These results were obtained through the use of equipment which gives more positive control of the heat supplied to the various rooms of the house.

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TABLE A. MONTHLY ENERGY CONSUMPTIONS FOR HOUSE HEATING FOR THE 1928-29 AND 1929-30 HEATING SEASONS

	Mean—Outdoor Temperatures F		Kilowatt-hours per Month	
	1928	*1929*	*1928*	*1929*
September	53.5	60.6	1,560	1,200
October	53.5	53.1	2,580	2,940
November	42.2	36.3	4,440	4,380
December	32.2	28.6	5,258	5,400
	1929	*1930*	*1929*	*1930*
January	17.6	20.1	6,546	5,820
February	21.6	37.1	5,208	3,720
March	43.2	36.1	3,786	4,260
April	51.1	49.2	2,784	2,580
May	56.8	61.2	1,890	1,620
Avge	41.6	42.5	Total 34,050	31,920

TABLE B. COMPARISON OF FIRST 4 MONTHS OF EACH YEAR (TABLE A) WITH MONTHLY AVERAGE WEATHER BUREAU TEMPERATURE RECORDS FOR CHICAGO

	1928		1929		1930	
	"Hot Room" Cons. Kw-hr	Mean Outdoor Temp F	"Hot Room" Cons. Kw-hr	Mean Outdoor Temp F	"Hot Room" Cons. Kw-hr	Mean Outdoor Temp F
September	1,560	53.5	1,200	60.6	913	64.0
October	2,580	53.5	2,940	53.1	2,724	51.6
November	4,440	42.2	4,380	36.3	3,401	42.4
December	5,258	32.2	5,400	28.6	4,679	29.7
Total	13,838	Avg 46.1	13,920	44.7	11,717	46.9

After a study of our experience in off-peak electric heating of houses, we reached the following conclusions: (1) the system is reliable; (2) it is positive in supplying heat; (3) it results in better living conditions in the home due to improved humidity (records show an average of 50 per cent relative humidity); (4) it eliminates fire risk, and (5) it can be made the most flexible system of house heating.

We realize that the rate for off-peak electric service is an important factor in the general realization of Dr. Steinmetz's prediction. At a rate of 1¢ per kilowatt-hour, electric heat is an expensive heat source when heating *becomes more than removing evening chill.* This should therefore suggest to the manufacturers of such heating systems and the users, as well, the fact that the heating plant should give the utmost in flexibility and economy of heat used in making a favorable comparison with fuel heat.

It is imperative, therefore, that the building be designed for minimum heat losses. A saving of 10,000 kwhr per year at 1¢ per kilowatt-hour would pay fixed charges on a considerable investment in heat conservation and distributing equipment. A home heated by electricity should be well insulated, have storm windows, and possibly weather strips. In addition, why not design the heat distributing system to deliver heat to various parts of the house when needed? I believe that with such a system it would be possible, on a rate of 1¢ per kilowatt-hour, to heat homes by electricity and have the 10-yr operating cost compare favorably with any fuel under similar operating and living conditions.

I congratulate Mr. Harrington on the thoroughness with which he has covered this subject and thank him for requesting this discussion. Our company realizes that the work we are doing now is a few years ahead of more general use of electric house heating but it hopes that our limited experience will add to the rapidly increasing data on this subject and the progress of off-peak electric house heating.

A. R. STEVENSON, JR. (WRITTEN): Being closely associated with Mr. Harrington I cannot add much by way of discussion, except perhaps to call attention to L. L. Elden's paper, presented to the New England Division of the *National Electric Light Association*, at Swampscott, Mass., Oct. 1, 1930, entitled Load Building at Night and Off-Peak Water Heating, which gives the views of a prominent public utility man on the economic advisability of off-peak rates.

P. S. LYON (WRITTEN): The author is to be complimented on his excellent presentation of this subject. I believe this is the first paper on the subject to be given to the Society. The scope of the paper is so complete that there is not much that can be added in the time allotted for a discussion.

Heating Viewpoint on Off-Peak Electric System

The paper demonstrates the entire practicability of the off-peak electric storage system. As a matter of fact, the heat distribution methods used to date have been radiation or warm air duct systems with which we are all familiar. Therefore, from the heating standpoint, heating engineers will see no mystery in this type of equipment.

Initial conversations with heating engineers and laymen, indicate general approval of electric heating as being a most practical, convenient, clean and automatic form; and, in fact, desirable from every standpoint, if the operating cost can be made reasonable.

Electric heating has been made reasonable in operating cost by the application of the off-peak principle. The paper states that reasonable off-peak electric rates are being made available. Certainly, the many advantages of electric heating eliminate the necessity that it should compete with other fuels solely on a price basis. Everyone is interested, however, in an approximate comparison. It can be said that electricity at ¾¢ per kilowatt-hour will compete with manufactured gas at 80 to 85¢ per 1,000 cu ft. The chief questions in the minds of most engineers are the feasibility of off-peak load to the utilities, and the practicability of their granting reasonable rates. It may be well to attempt to assist in answering these questions by pointing out the principal things to be considered.

Off-Peak Hours

Off-peak hours for an electric utility system may be defined as the hours out of the 24 during which the existing load is considerably lower than the maximum load for the 24 hr.

A discussion here, of course, must be general and illustrative only. The principle points can best be understood by reference to a daily load curve, such as shown in Fig. A. This is a composite curve for an electric system serving an eastern territory, which includes a large industrial city. The daily system peak is 109,500 kw, occurring at 6:00 P.M. There is a secondary peak occurring just before noon. The analysis of the system and the characteristics of the community indicate that the peak periods cover the hours of 9:00 A.M. to 12:30 P.M. and 3:30 P.M. to 6:30 P.M. The off-peak hours selected for this system total 17½ out of the 24 hr.

A rather common and limited conception has restricted the term "off-peak" to night hours. However, this illustrative load curve indicates that many daytime and evening hours are also available for which a utility company could accept off-peak business. In making off-peak energy available at low rates during certain daytime as well as night time hours, it is intended to make use of facilities, otherwise used only to a small percentage of capacity during those hours. A successful application of this principle by a utility company has been mentioned by the author in referring to the telegraph companies. A night letter may actually be sent over the wire in the middle of the afternoon, although not delivered until the next morning; the message has been transmitted at a time when the facilities were not otherwise used.

It is a quite obvious fact that longer charging hours permit smaller tanks and lower connected load, which, of course, make lower-priced heating equipment possible. The utility company gains by longer hours in obtaining more potential heating customers due to less expensive heating equipment. A heating customer takes the same quantity of kilowatt-hours for the year whether charging periods are short or long. Therefore, longer charging hours make for a larger increase in business for the utility because more heating customers are obtainable.

The daily load curve (Fig. A) shows that 15,000 kw of connected off-peak load

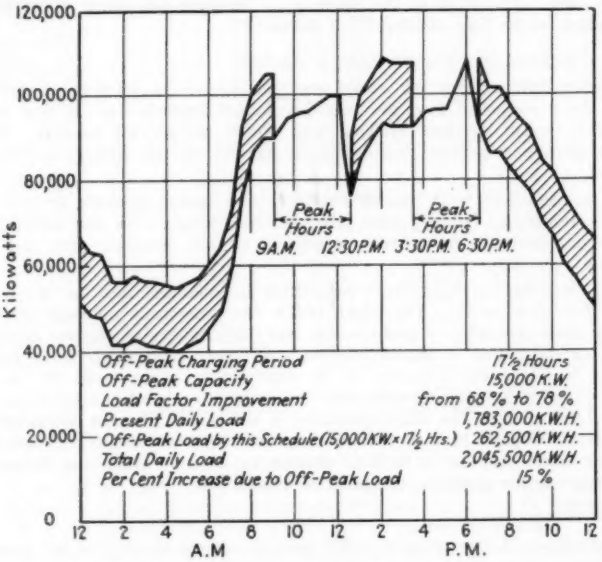

FIG. A. DAILY LOAD CURVE FOR ELECTRIC SYSTEM

are available for the hours selected. In making the study, the utility engineers would not try to set up the greatest total off-peak load by selecting short hours. The characteristics of the community should determine the selection of off-peak hours in relation to total off-peak capacity to take care of prospective off-peak heating business. In the illustration, 15,000 kw will quite probably take care of prospective heating customers for many years to come, since this is an industrial community.

Utility Company Standpoint

Probably no industry, as a whole, is seeking new business more aggressively than the electrical industry. A group of off-peak heating loads, obtained during the hours and at the rates determined by proper analysis, should give the electric company one of the best and most profitable forms of increased business. Probably the most attractive feature is the fact that this load automatically stays off the peaks, and that, therefore, no major investments are necessary because it makes use of otherwise unused facilities. If the rates are properly set, they will attract sufficient customers at a profitable total net return to the company. It should be kept in mind

that the company could not obtain building heating business except by setting up off-peak rates.

A utility is attracted to off-peak business by the potential improvement of power factor. This heating load is ideal to the operating man because it has unity power factor, which will tend to improve the system power factor.

In determining off-peak heating hours and off-peak electric rates for heating, a utility company must consider not only the system as a whole, but also the generating stations, the high tension transmission system, the substations, and the local distribution. Any one of these parts of the system might influence the selection of hours more than the complete system by itself.

Regarding determination of rates, it is evident that, in general, additional investment for generating and transmission equipment is entirely unnecessary. Usually, no additional investment will be necessary for substation or feeders. In taking on an off-peak heating customer, the electric company's additional expense, in general, includes only a larger customer transformer and heavier leads to the building. The prospective heating customer is already a user of electricity, so that added expense for meter reading and for billing and accounting is small. Therefore, the utility company's costs for this off-peak service consist chiefly of fuel, some small additional labor and maintenance expense at the generating plant, and certain investment charges on the increased size of customer transformer and service. There is no intention, and in fact, no necessity, to discuss here the theories of rate-making.

Many leaders of the electric industry are looking to residential loads to be the big opportunity for future increase in business. From *Electrical World* we learn that in August, 1930, 660 kwhr was the average yearly consumption per residential customer, while 4 years previously, it had been 376 kwhr. No other class of customer offers so large a field for increasing electric business as the off-peak heating field and it can be seen to fit into the visions of many of the electric utility leaders.

Probably the large majority of people, including heating engineers, have not taken the time to think along the lines of the advantages of off-peak heating load to the electric industry. I have tried to outline a few of the features from the electric company's standpoint so that we, as heating engineers, may realize that the electric industry can well afford, and will undoubtedly rapidly proceed, to set up the reasonable rates that were pointed out by the author of the paper. If we foresee that relatively low electric rates are economically sound, heating engineers will realize that the only apparent disadvantage of off-peak electric heating has been removed, and that this system of heating should progress rapidly in this country. It is interesting to note that in Europe the use of off-peak electrical storage heating systems has been increasing for many years.

LEE P. HYNES (WRITTEN): No one who is familiar with power conditions doubts the desirability of securing as uniform a load demand as possible, but there is considerable diversity of opinion as to the best method of accomplishing this desirable result, and I would like to point out some important factors which have not been referred to in Mr. Harrington's paper.

I agree with him that electric heating does offer one of the principal opportunities for improving load conditions, but I do want to point out some factors which must be carefully considered, as electric heating has always suffered at the hands of its over-enthusiastic friends.

I firmly believe that except in isolated cases, it is not practical to heat homes by electricity. I do not mean that a home cannot be heated by electricity, as I have personally seen homes satisfactorily heated by that method under particularly favorable conditions, but I am sure there is no possibility of any extended use of electricity for this purpose.

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Wherever it has been successfully used, extreme care has been taken to insulate against heat losses and proportional savings would be obtained with other heating mediums with similar care, so in the last analysis, the problem resolves itself into relative cost per unit of heat delivered usefully at the radiators. Many power companies have made temporary experimental rates of from 1 to 1½¢ per kilowatt-hour, but it appears to me extremely doubtful whether current can be sold for less than 1¢ per kilowatt-hour under the conditions required for house heating service and, as a rate of 1¢ per kilowatt-hour is equivalent to about $40.00 per ton for coal, it appears logical that some form of fuel heating with either liquid, gaseous, or pulverized fuel under automatic control, would always show greater economies than electricity. It is hardly conceivable that more than a mere fractional per cent of homes in any community could be heated electrically, regardless of the rate.

In saying this, I have in mind primarily the power conditions and their relation to the particular requirements of house heating. It is true that off-peak operation would help meet this situation to a limited extent, but we must remember that even with off-peak heating we must work on a 24-hr cycle and must have an installed capacity in each house capable of storing during certain hours of the day, all the heat required for the coldest day in the year.

Unfortunately the average power system has only about half as much off-peak power available in winter as in summer. Inasmuch as we are limited in available power for diversion for house heating to the maximum off-peak power, which the electric company can spare on the coldest day of the year, it will be obvious that off-peak house heating can utilize only a relatively small percentage of the possible off-peak power throughout the year. Furthermore, inasmuch as electric heating will always be more expensive than fuel heating, both in installation cost and in operating cost, the people who can be induced to use it will be scattered throughout the community rather than located near the powerhouse. This will mean the installation of larger service lines and transformers wherever a house is to be heated and this will usually cost the householder from $200 to $300.

It is doubtful whether any power company can so accurately determine its available off-peak power or its cost in the future, that it would guarantee any rate to a householder. A householder making an installation in which he eliminates chimneys, etc., as Mr. Harrington predicts, has a right to know what his power rate will be for a considerable period of time.

Recently an eminent consulting engineer for one of the leading power companies in this country, in a technical article on off-peak power rates, pointed out the advantages of utilizing this load as far as possible, but warned power company officials when he said:

> Off-peak service may be supplied under absolute control of the utility, both as to time and quantity used. Either of these factors may be changed at will to suit operating conditions and customers' requirements.

Off-peak storage electric heating has long been used in Switzerland, Sweden, and other European countries having an abundance of surplus water power and high fuel costs, and in recent years a considerable use has developed in England, but in England they do not favor house heating as the best way of utilizing off-peak power. They generally favor installations in large buildings or industrial plants having heavy demand for current for purposes of power or lighting during part of the day, so that it is possible to utilize the current during other periods for storing up heat energy. In England they have favorable winter load conditions which are met in this country in few large industrial centers. In this way, no additional lines and meters are required and it is possible to offer particularly favorable rates because of the large total amount of power taken by one consumer. In my judgment, this is the logical way to utilize available off-peak power, rather than for heating of individual homes.

However, even under these favorable industrial circumstances, it is seldom logical to do general heating by electricity where a fuel system will do the job equally well.

There are, of course, many special cases which justify the use of electricity. For instance, off-peak electric heaters can be tied-in with fuel systems as auxiliaries or as boosters so that surplus power can be utilized whenever desirable. A good example of temporary heating by electricity is at the Ford Motor Company's plant, Green Island, N. Y., where a 3,600-kw electric hot water boiler was installed to heat the factory until the full output of the hydro-electric plant on the Hudson River will be needed for other power purposes through growth of the business. At such time a fuel boiler will simply replace the electric boiler, as the current is more valuable for other purposes than for heating.

It has been inferred that pumped circulation is particularly a feature of electric water heating, but the Ford installation will utilize pumped circulation for both fuel and electric heating, so that again we find electric heating resolves itself to a purely economic basis of comparative Btu cost.

There are many industrial processes where off-peak heating could be utilized and power companies have not yet given sufficient study to this outlet for their surplus off-peak power. This problem is being given careful consideration by power engineers and operating officials, but I am sure the consensus of opinion is that we must go forward with extreme caution and only in accord with results definitely proved.

One thing we must not lose sight of is that off-peak heating no matter how carefully done, is wasteful of Btu. This may be more than off-set by the lower cost of off-peak current as compared to full-time service, but the fact remains that there is a substantial amount of heat wasted during standby periods. This percentage is especially high during mild weather when the quantity of heat needed is relatively small because the standby loss is almost a constant figure.

Another unfortunate fact is that we cannot extract from the system the heat we put into it to build up the temperature to the lowest permissible temperature at which we can use water in the radiators. We must have storage sufficient to allow us to do all the heating with water above the temperature of 125 F and this makes storage heating somewhat unsatisfactory for intermittent service.

A recent example of this came to my attention where a small industrial building had a storage heating system installed of 90 kw capacity and during a month of mild weather had standby losses greater than the total need for useful heat. As a result of this experience, the management cut off the storage system and installed half the capacity of heaters in local units under thermostatic control, and cut down the monthly demand for current to nearly one-half the result obtained with storage heating.

This was not the fault of storage heating. It was due to its installation under circumstances where it was not practical, and I only cite it to emphasize that storage heating is not a panacea for all heating troubles, nor for all power sales troubles, but that like most other engineering problems it must be carefully worked out. It is not nearly so much a commercial selling problem as our other types of heating.

The second part of Mr. Harrington's paper deals with electric water heating for domestic supply. This I feel is a much more fertile field than house heating as a means for improving the load curves of power companies.

Here again there is great diversity of opinion as to whether true off-peak heating under the control of a time clock is advisable and there are two accepted methods of doing this even where it is agreed that time clock control is advisable; namely, by the use of small tanks and high temperature with mixing valves, or large tanks and moderate temperatures.

It is true that larger tanks have less radiating surface in proportion to their volume. Hence the radiation losses are greater for the small tank with the higher temperature

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water. This is proved in practical service by the fact that a 60 gal heater has only 50 per cent greater loss than a 30 gal heater under similar temperature conditions.

Inasmuch as high-temperature water involves the use of rather complicated mixing valves to mix cold water with the hot water before using, and greatly increases corrosion and scaling troubles (especially with some waters), I think there is serious doubt whether these disadvantages will compensate for the slight increased first cost of a larger tank.

However, this is just a detail aside from the main question regarding the value of water heating as the most satisfactory power load. Examination shows that it is far superior in every way to the house heating we have just discussed. This is true for several reasons.

First, the demand for hot water exists throughout the whole year and has no direct relationship to the weather. In fact, a properly designed system will use much less electrical energy in the winter than in the summer, which is a most desirable feature and exactly the reverse of house heating.

An electric hot water system in a home should utilize some of the heat of the fuel heating system for the house to pre-heat the water before it enters the electric heater. I have had my own house equipped this way for several years and it reduces the electric power demand in the winter to approximately one-half that of the summer time, and the colder the weather, the lower the power demand. This exactly parallels the ideal load condition for the power company, as available off-peak power drops rapidly in the winter time.

The second big advantage of domestic water heating is that the individual load is small, requiring no increase in power service to the house. Hot water for the average home can be supplied by a properly designed heater of from 1 to 2 kw capacity, which does not place any burden on existing wiring. Furthermore, it does not mean any more change in house construction than the installation of an electric refrigerator or radio.

An electric water heater does not cost appreciably more than an equally well built gas heater. Its operating cost at rates now available in many places is not appreciably greater, and as every home is a potential user of a moderate amount of electricity for heating water, this offers a much more logical outlet for all possible surplus power than any other means.

Another factor which is favorable in water heating is diversity. People do not all require hot water at the same time. Even where no time clocks or other special control are installed to insure true off-peak heating, about 60 per cent of the total use of current will fall during off-peak hours, largely due to the considerable use of hot water at night for bathing which automatically brings the heater into service to replenish the supply before morning.

It has been found on a large power system with many 1,500-w, 30-gal water heaters, diversity alone will cut the total demand at one time to half the connected load and that each heater can supply at least 90 gal of water per day under the average conditions in which it is used and at 110 F temperature rise.

If true off-peak heating with time clock control is used, the connected load of each water heater must be practically double what would be required for non-controlled heaters, so as to insure storing a full 24-hr supply between midnight and morning. Eight hours is the maximum off-peak permissible because of the heavy draw of water which may come late at night for bathing. Therefore, the connected load per heater for the system I have just mentioned, would have to be 3,000 w per heater with 90-gal storage for true off-peak heating, as against 30-gal storage with 1,500-w heater as installed on approximately a 60 per cent off-peak schedule without time clocks.

Under these conditions, it will not pay every power company to install time clocks as these cost from $25.00 to $40.00 per heater. Any time clock to be satisfactory must be of the electrically-wound spring type. Straight synchronous clocks are not satisfactory because in case of any temporary interruption of the service, even by so simple a cause as a blown fuse, they will lose time and get out of cycle with the peak periods.

I recently talked with a power company official who had been responsible for the servicing of 1,000 time clocks on his lines and who was not enthusiastic about this burden. However, I am satisfied that there are now several satisfactory clocks on the market and on some power lines their use would be entirely justified. Thus it will be seen that a careful analysis of the particular power conditions of each utility company must determine whether full off-peak service is warranted.

I hope no one will misunderstand my attitude in commenting on Mr. Harrington's paper. I am glad indeed to see some slight recognition being given electric heating by this Society. I have been a member of the Society for about 12 years and as far as I recall, this is only the second paper on electric heating which has ever been presented at an Annual Meeting.

I do, however, urge that heating engineers approach carefully this problem of electric heating as an installation can be either eminently satisfactory or totally disappointing in service. It is not safe to deal with generalities, but each case must be carefully studied from both a thermal standpoint and a power load standpoint. A failure to consider both these essential factors will certainly lead to disappointing results.

PROF. E. O. EASTWOOD: I may be able to throw some light on one phase of this paper, inasmuch as I come from one of the states where hydro-electric development is well under way. The cities of Tacoma and Seattle own and operate hydro-electric plants and 4 or 5 years ago the city of Tacoma was selling electricity for heating purposes at the rate of ½¢ per kilowatt-hour. They have not been able to maintain that rate. There was considerable development of household heating at that time, due to the low rate, but the city has been obliged to withdraw the rate so that the increase of domestic heating by electricity has, in my opinion, ceased. The rate in Seattle for domestic purposes varies from about 5½¢ per kilowatt-hour to 1¢ operating on a sliding scale. Last summer I installed an oil burner in my house. I find that the cost of the fuel will be about two-thirds the cost of coal. I found by previous investigation that it would cost about 100 per cent more to heat with gas at the prevailing rate in that city. I mention these things because I think if electricity increases in popularity it will be done more on the basis of cost than on the basis of these other attainments which have been enumerated in the paper.

The average person is more concerned with the cost of fuel than he is in the advantages of a system. So what I have said to you would emphasize the importance of the paper, inasmuch as it appears that at least at this time the use of electricity for household heating will depend upon some system of using the off-peak load.

ELLIOTT HARRINGTON: Five per cent, as mentioned by Mr. Haar, is a low interest for borrowed money in making an economic study. As one of the electrical engineers for the Board of Transportation of the City of New York, Mr. Haar in making his studies is using an interest rate which a municipality might have to pay on its bonds. We understand his study was on the heating of sheds, sub-stations and shops for one of the new municipal subways in New York City.

Mr. Waby states that the system in the Mitten residence is a typical Hall system. As a matter of fact, the Hall company has never installed a system of this type. The one in the Mitten house is similar to the one Mr. Hall installed in his own residence at Fort Wayne, while the Hall system was still in the developmental stage.

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It consists of a well-insulated hot room with water storage tanks around which the air circulates, to be heated before it passes up through ducts to the house.

Mr. Waby's statement that the system is reliable, positive in supplying heat, and results in good living conditions, is definitely corroborated by experience in residences of various types in other territories under varying climatic conditions, and with the storage equipment supplying various types of radiation systems.

It is hardly for us to question the statements made by Mr. Waby. However, there is no real reason why the humidity in the electrically heated house should be any better than in a house heated from some other source. In a report on the Mitten installation, we find that the basement floor was at least damp a part of the time. The air was recirculated from the house through the basement into the hot room, and then back through the house; it is evident that the air in the house would pick up some moisture from the wet basement floor.

According to Mr. Waby, a rate of 1¢ makes expensive heating. It might be in order to again point out that lower rates are entirely feasible from the utility company's standpoint.

We agree with Mr. Waby that it is desirable to have the heat distribution system deliver heat to the different parts of the building only when it is needed in those parts. We, like other people, have been and are now thoroughly investigating this. Installations have been made which give separate controls for different parts of the building.

As an added point to those presented by Mr. Lyon regarding the desirability of this type of load for the public utilities, there has been in the public and technical press, within the last several months, a continued reference to the stability and persistence of the domestic load of the electric light and power companies as compared to the industrial load, which decreased quite markedly as a consequence of the widespread national decrease in industrial activity. The development, therefore, of added uses of electricity in the home would undoubtedly be looked upon favorably by the utilities as a load of stable and permanent characteristics, which might be expected to persist at an average rate and not decrease with any decrease in industrial activity.

It might help to point out that the average water heater customer will take 5,000 kwhr per year, and the average house heating customer will take 50,000 kwhr per year, as compared to the figure of 660 kwhr for the average residential customer today.

Mr. Hynes is positive in his statement that electric heating for homes is impractical. This point is argumentative, and possibly the statement should be challenged. He indicates that in all houses where electric heating has been done, insulation of the houses has been undertaken. We know of a number of houses where no insulating was done because of the installation of electric heat.

Mr. Hynes says that 1¢ electricity is the equivalent of $40.00 coal. We believe the comparison is much closer to $30.00 coal than to $40.00 coal.

He also states that certain storage heating requirements must be met. As a matter of fact, those are exactly the requirements which these systems are meeting.

Although, as Mr. Hynes points out, a power system has only half as much off-peak power available in the winter as in the summer, it is quite likely that that half may be sufficient to heat all of the prospective electrically heated houses for some time to come.

We do not believe the utility companies are, in general, going to charge the householder for the transformer and service equipment as indicated by Mr. Hynes. There are going to be some cases where long runs are necessary and may have to be charged, but when a utility company sets out to get the off-peak heating load, it

looks as though they would undertake the investment for connecting to the lines themselves.

Mr. Hynes questioned the guaranteeing of any rate by a utility company. It seems that the power companies will be able to guarantee one set of charging hours to a certain block of customers, and then change to other charging hours for a subsequent block of customers. This would seem to answer Mr. Hynes' argument. We do not believe the quotation given by Mr. Hynes means that the power company will change the conditions for any existing off-peak customer, but that they will, of necessity, have to, from time to time, take additional blocks of load on different hours.

In regard to the use of a pump with the electric hot water heating system, it has never been our intention to infer that this was something new. We actually have systems operating without pumps.

Mr. Hynes is apparently familiar with utility rates and the potentialities of the sale of energy under various forms of rates. He has referred to the possibilities in industrial processes where off-peak heating could be utilized. He has not, however, developed the complete value of this point. There are at the present time, and there have been for many years, in most of the territories in this country, rates available to large power users called "maximum demand rates." Briefly, these rates permit the purchase of electricity from a power company on the basis of two charges. The first charge, called the "maximum demand charge," is based on the maximum kilowatt demand placed by the customer against the power company's lines. This demand charge is effective for certain restricted hours out of the 24, and sometimes for certain restricted months out of the year. Outside of these restricted hours and restricted months, the customer has available the use of electric energy at low energy rates indeed. These rates for energy may vary from 4 to 7 mils per kilowatt-hour, depending upon the locality, the application, and the type of contract. Electric energy at 4 mils becomes competitive on a building heating basis with manufactured gas at about 45¢ per 1,000 cu ft, and an efficiency of 70 per cent, and compares with anthracite coal at about $13.00 a ton, on a basis of 40 per cent efficiency. This field for the application of heating equipment is already here, and carries no barrier from the standpoint of rate, because this type of rate has been in operation for many years.

Regarding the storage heating system installation mentioned by Mr. Hynes, he undoubtedly refers to two installations which were made by the American Gas and Electric Company at substations in Marion and Muncie, Ind. The trouble was not with the storage equipment, but with the fact that no one knew how much waste heat would be available from the synchronous condensers. The American Gas and Electric Company engineers furnished a figure for heat which would be available from this source. As a matter of fact much more heat was obtained so that the heating system was actually larger than necessary. It meant that the percentage of heat dissipated from tanks was much larger than had been expected.

Mr. Hynes shows that he has missed the point in the off-peak principle when he states that even without time control, 60 per cent of the total use of current with water heaters will fall during off-peak hours. Even though only a fraction of 1 per cent of the energy is taken during peak hours, the utility company has to provide the full facilities which are used for only a short time.

Mr. Hynes creates an entirely incorrect picture when he mentions 8 hr as the maximum off-peak permissible. A utility which limits its off-peak water heating load to 8 hr is the exception. They are setting up 13, 15, 17 and even 20 hr.

Mr. Hynes refers to the cost of time switches for controlling water heaters. His figures of $25.00 to $40.00 per time switch undoubtedly refer to spring-wound devices. Synchronous time switches, extremely simple and rugged in construction, and using the synchronous element which has been in service for other purposes for several

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years, can be purchased for $15.00 or less. While the synchronous device might be subject to an occasional temporary outage, corresponding to the outage in the electric service to the premises, this would not affect the control of the proper supply of hot water to the premises, as the total cycle for the switch would continue, but would merely be displaced by the length of time of the outage. If the outage was merely local, and not widespread in the territory, it would not affect many time switches, and would be of no moment to the electric light and power company. If it was a serious and prolonged outage of an hour or more, this would certainly be known to the utility, and the time switches in the territory could be promptly restored to the proper setting. Time switches of this type are made with a visual inspection means to permit easy check of their accuracy of setting.

Professor Eastwood's discussion is interesting and valuable, but it should be understood that his reference to the rate of ½¢ per kilowatt-hour, which existed in Tacoma, refers to an unrestricted rate—that is, electricity to be used at any time during the 24 hr. On this basis, the rate is, by present standards, somewhat low. It might be possible, however, to profitably sell energy at such a rate if the apparatus was properly designed for off-peak operation and some form of off-peak control installed.

No. 897

HEATING WITH EXHAUST STEAM

By PERRY WEST,[1] NEWARK, N. J.
MEMBER

WASTE heat should be used for heating and ventilating requirements wherever it is available. The sources of such waste heat include electric or power generation, industrial and condensing processes, and furnace gases.

Where heat and power or electric current are required, steam may first be used for generating the power or electric current and the exhaust from the generating units then used for heating. Likewise in many industrial processes requiring heat at certain temperatures, steam may first be passed through engines or turbines, for generating any power or electric current required and used for the processes at the pressure required to give the desired temperature.

The exhaust steam may be circulated directly through the radiators, heaters, air heaters, humidifiers or other elements of a heating, or of a heating and ventilating system, or it may be used for heating water which in turn may be circulated through similar elements of a hot water heating system. It may also be used for domestic water heating.

Frequently there is a fluctuating load on the power or electric generating apparatus which in turn produces fluctuating quantities of exhaust steam. The quantity of steam required for heating is also variable. These two variables do not, as a rule, coincide so that the total quantity of exhaust steam produced may be either equal to, a certain percentage less or a certain percentage more than the total quantity of steam required for heating, or for heating and ventilating either during the heating season or throughout the year. These relations change throughout the day and night and throughout the seasons of the year.

There are also wide variations in the heating requirements due to changes in weather conditions and wide variations in the exhaust produced due to such changes as in the demands for lighting on dark or light days, for elevator power during rush or slack periods, for air conditioning on good or bad days and for many other variables in the demands for power or light, depending on the character of building and its operating requirements.

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[1] Consulting engineer.

Presented at the 37th Annual Meeting of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, Pittsburgh, Pa., January, 1931, by P. D. Close.

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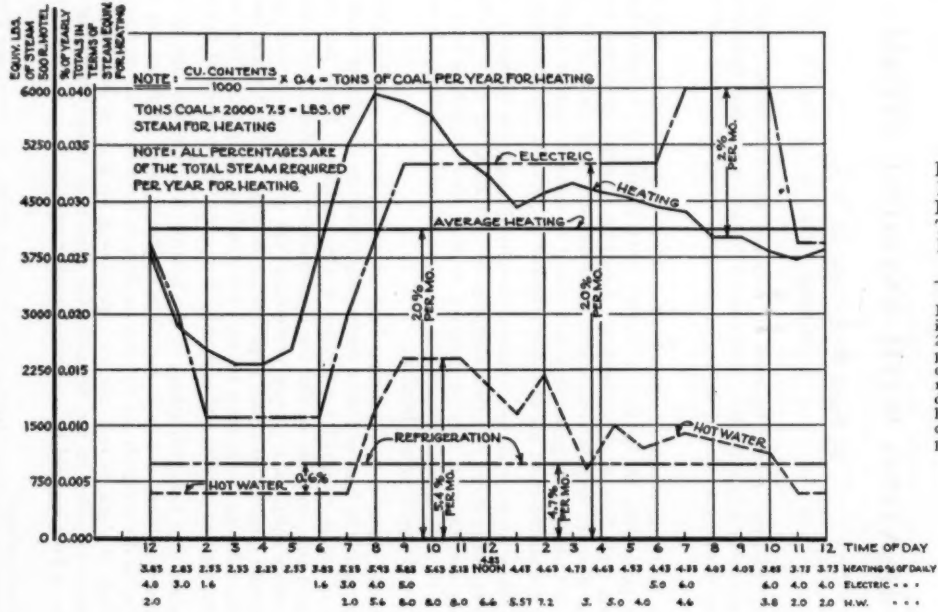
EQUIV. LBS. OF STEAM 500 R. HOTEL
% OF YEARLY TOTALS IN TERMS OF STEAM EQUIV. FOR HEATING
NOTE: CU. CONTENTS / 1000 X 0.4 = TONS OF COAL PER YEAR FOR HEATING
TONS COAL X 2000 X 7.5 = LBS. OF STEAM FOR HEATING
NOTE: ALL PERCENTAGES ARE OF THE TOTAL STEAM REQUIRED PER YEAR FOR HEATING
ELECTRIC
HEATING
AVERAGE HEATING
20% PER MO.
20% PER MO.
2% PER MO.
REFRIGERATION
HOT WATER
HOT WATER
0.6%
5.4 % PER MO.
4.7% PER MO.
6000 0.040
5250 0.035
4500 0.030
3750 0.025
3000 0.020
2250 0.015
1500 0.010
750 0.005
0 0.000
12 1 2 3 4 5 6 7 8 9 10 11 12 1 2 3 4 5 6 7 8 9 10 11 12 TIME OF DAY
NOON
HEATING % OF DAILY
ELECTRIC
H.W.

FIG. 1. JANUARY AND FEBRUARY HOTEL LOAD CURVES IN TERMS OF STEAM EQUIVALENTS FOR HEATING [a]

[a] Ventilation Steam Requirements (15%) not included. Generators at 25 lb per indicated horsepower per hour; 30 tons of refrigeration for 500 rooms at 25 lb per indicated horsepower per hour. No allowance for condensation (about 20 per cent).

ADJUSTMENT BETWEEN HEATING LOADS AND EXHAUST STEAM

The way in which the quantities of exhaust steam produced and the quantities of steam required during the various intervals of time throughout the year coincide will determine how much may be used, how much must be thrown away and how much live steam must be used to supplement the exhaust steam deficit, unless some means is adopted for adjusting the supply to the demand.

One method of making this adjustment is to store exhaust steam or its equivalent in heat, when it is in excess of the demand, and feed this to the system when the exhaust being produced is below the demand. Hot water accumulators may be used for this purpose by storing heat in water to be fed to the boilers, in water to be used for domestic hot water service, in water to be used in a hot water heating system, in water to be used for process work, or for some combination of these purposes.

Another method is to adjust the quantity of exhaust produced somewhat in concordance with the heating demand. A bleeder turbine from which the exhaust may be taken for the lower heating demands and additional higher pressure steam from its successive stages as this demand goes up is frequently a satisfactory solution for a case where the normal exhaust is deficient. If such a turbine is also arranged to run condensing under various controllable degrees of vacuum and arrangements are made for either circulating the exhaust through the heating system under various pressures above and below atmospheric pressure or for similarly using the exhaust for heating water, such an arrangement may adjust the exhaust to a wide variation of heating demands.

With such an arrangement, the exhaust supply may be automatically adjusted to the demand by having automatic control of the vacuum in the turbine from an outdoor thermostat operating on the flow of cooling water on the condenser and remote control valves in the bleeder steam supplies controlled from the same or a coordinate thermostat. This arrangement has certain advantages over that in which exhaust steam is discharged to the atmosphere when it is in excess of the demand and where live steam at boiler pressure is used to make up the deficit when the exhaust is insufficient. For example, the condensing of the excess exhaust steam increases the efficiency of the turbine so that the total heat rejected to the condenser is less than would otherwise be thrown away, while the use of the bleeder steam conserves heat by allowing a certain amount of power to be generated from this steam before it is fed to the heating system.

A modification of this method of adjustment, especially as between the heating and non-heating seasons, is to use engines or turbines which may be run condensing throughout the non-heating season and non-condensing throughout the heating season. Another modification is to have non-condensing engines or turbines for use during the heating season and low pressure exhaust condensing turbines for using the exhaust steam from these during the non-heating season. This latter arrangement affords a high efficiency since each unit operates under conditions for which it may be best designed. The low pressure turbine in this case may even be made a bleeder turbine and be arranged to operate on various degrees of vacuum with automatic control as previously stated. While this arrangement of units generally shows a high overall economy there is some disadvantage in installing condensing units

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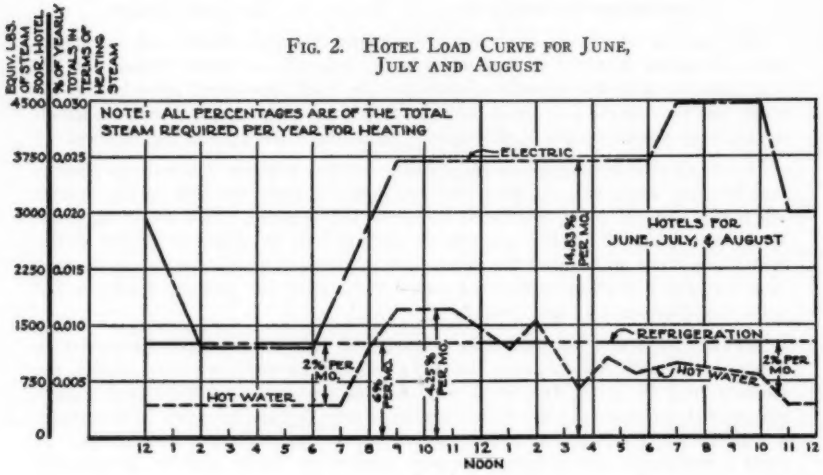
EQUIV. LBS. OF STEAM 500 B. HOTEL
% OF YEARLY TOTALS IN TERMS OF HEATING STEAM
4500 0.030
3750 0.025
3000 0.020
2250 0.015
1500 0.010
750 0.005
0
NOTE: ALL PERCENTAGES ARE OF THE TOTAL STEAM REQUIRED PER YEAR FOR HEATING
ELECTRIC
HOTELS FOR JUNE, JULY, & AUGUST
14.63 % PER MO.
REFRIGERATION
HOT WATER
2% PER MO.
4% PER MO.
4.25 % PER MO.
2% PER MO.
HOT WATER
12 1 2 3 4 5 6 7 8 9 10 11 12 1 2 3 4 5 6 7 8 9 10 11 12
NOON

FIG. 2. HOTEL LOAD CURVE FOR JUNE, JULY AND AUGUST

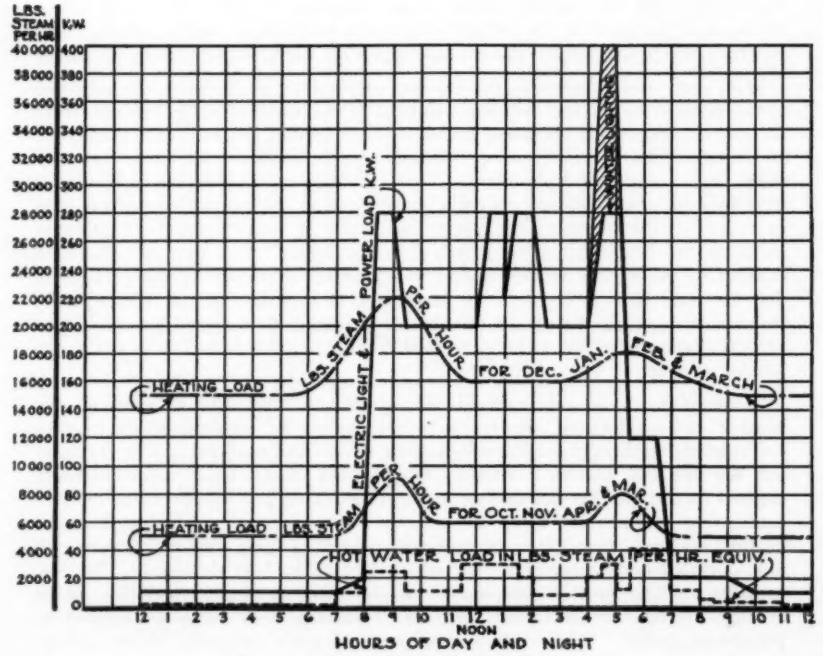

FIG. 3. OFFICE BUILDING LOAD CURVES

which have to be frequently started up or shut down and by-passed and hence a straight bleeder turbine is frequently used as a matter of simplicity.

Still another method is to adjust the heating demand so that it will balance as nearly as possible the exhaust steam produced. The use of hand operated or automatic remote-control sectional, or zone, regulating valves for separately controlling the various sides and vertical sections of a building in accordance with the exposures, wind velocities, sunshine, etc., the use of automatic temperature control, fractional distribution of steam, distribution of steam under varying degrees of vacuum and the use of forced hot water with temperature control, all tend to keep the demand commensurate with the actual weather requirements, thereby reducing the maximum demand and at the same time the amount of fluctuation by supplying heat only where it is needed.

HEATING LOAD AND AMOUNT OF EXHAUST STEAM AVAILABLE

In all of these adjustments the question of relative economies and the cost to produce them must necessarily enter into the problem. The efficiencies of engines and turbines vary with the types, sizes, speeds, load conditions and with the steam pressure, temperature, back pressure or vacuum under which they operate. Actual guaranteed water rates for the units and conditions to be employed on any particular operation should be obtained from several reputable makers before endeavoring to estimate the exhaust.

Load conditions and heating requirements must then be computed for the various hours of the day and night and for the various days of the year. Generally, a 24-hour day load and heating requirement chart for each of the four seasons of the year will be sufficient although in many cases the load may be different for some days of each week and of course over holidays and other non-operating periods.

Typical load curves are shown by Figs. 1, 2, 3, and 4. In the hotel diagrams (Figs. 1 and 2) it will be noted that the steam required for generating the refrigeration power is shown separately and is not included in the steam required to generate the remaining electric current or power. This is for the purpose of comparing it with the hot water load, which it practically equals in all of these charts. For this reason the refrigerating apparatus in this class of building may frequently be steam driven and the exhaust thus produced be used with good economy for heating the domestic hot water even where the remainder of the electric current is purchased from the outside and steam or hot water direct from the boiler plant is used for heating.

PROCEDURE OF MAKING ANALYSIS

In making an analysis of any particular operation it is necessary to develop charts similar to Figs. 1, 2, 3 and 4. The total steam required for heating per square foot of direct radiation, or its equivalent, per season, for different classes of buildings is as follows:[2]

For manufacturing or commercial loft type, where steam is used to heat the premises during the day hours to maintain 65 to 68 F from 9 A. M. to 5 P. M. No Sunday or holiday use and no night use. Factor: 325 lb. per square foot of radiation per season.

[2] See Chapter 24, A. S. H. V. E. GUIDE, 1931.

For office buildings using steam during daylight hours only to maintain 70 F from 9 A. M. to 6 P. M. for approximately 240 days (heating season). No night use. Factor: 400 lb. per square foot of radiation per season.

For office buildings using steam during day hours and at night when required to 7, 8 and 9 P. M. (customary where there are stock brokers or banking offices), 204 days. Factor: 500 lb. per square foot of radiation per season.

For residences of the block type (not detached) where high-class heating

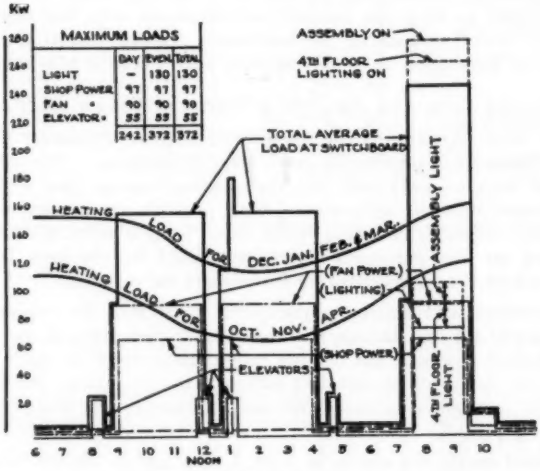

FIG. 4. HIGH SCHOOL LOAD CURVES

service is required somewhat similar to apartment buildings. Factor: 550 lb. per square foot of radiation per season.

For apartment houses where high-class heating service is required. (Steam off at midnight.) Factor: 650 lb. per square foot of radiation per season.

For hotels (commercial type) where very high-class service is required; 24-hour service. Factor: 800 lb. per square foot of radiation per season.

Of the total steam required for any particular service, the following percentages of the whole may be apportioned to the various months of the heating season in the colder climates: January, 20 per cent; February, 20 per cent; March, 15 per cent; April, 10 per cent; May, 5 per cent; October, 6 per cent; November, 8 per cent; December, 16 per cent.

To determine the amount of exhaust steam available, it is necessary first to develop electric current or power load curves and from these to estimate the amounts of steam used during various periods. Fig. 5 gives the ideal water rate for engines or turbines working under different initial pressure and back pressure or vacuum conditions, with corrections for superheat in the initial steam. Table 1 gives the factors by which ideal water rates may be multiplied to obtain average actual expected water rates for engines and turbines of

TABLE 1. APPROXIMATE FULL LOAD EFFICIENCIES FOR NON-CONDENSING ENGINE AND MULTI-STAGE TURBO-GENERATOR (GEARED TYPE)

Rated Capacity at Full Load Kw.	Mechanical Efficiencies of Engine Units	Mechanical Efficiencies of Turbine Units	Brake Potential Efficiency Ratio of Turbines	(a) Factors for Converting Ideal Water Rates to Actual Water Rates (for Turbines Only)	Generator Efficiencies at Full Load
100	0.900	0.930	0.45	2.22	0.932
200	0.905	0.935	0.50	2.00	0.934
300	0.910	0.940	0.54	1.85	0.936
400	0.915	0.942	0.58	1.74	0.938
500	0.920	0.945	0.61	1.64	0.940
600	0.925	0.947	0.64	1.56	0.942
750	0.930	0.950	0.65	1.54	0.944
1000	0.935	0.952	0.67	1.49	0.945
1250	0.940	0.955	0.67	1.49	0.947
1500	0.945	0.957	0.67	1.49	0.950
2000	0.950	0.960	0.68	1.45	0.954
		CONDENSING TURBINES			
500		0.950	0.68	1.45	0.940
600		0.952	0.69	1.44	0.942
750		0.954	0.70	1.43	0.944
1000		0.956	0.70	1.43	0.945
1250		0.958	0.70	1.43	0.947
1500		0.960	0.70	1.43	0.950
2000		0.960	0.70	1.43	0.954

	(For Non-Condensing Engines only)
Single valve automatic engine units up to 400 kw	1.64
Four valve automatic engine units up to 400 kw	1.42
Unaflow engine units up to 400 kw	1.20

[a] These factors are for indicated engine or brake turbine loads and do not take into account the mechanical efficiency of engine units or generator efficiencies for either engine or turbine units; but do include the mechanical efficiency of turbines.

TABLE 2. FACTORS BY WHICH TO MULTIPLY ACTUAL WATER RATES AT FULL LOAD TO OBTAIN ESTIMATED WATER RATES AT FRACTIONAL LOADS ON NON-CONDENSING ENGINE AND TURBINE DRIVEN ELECTRIC GENERATING UNITS [a]

Size and Kind of Unit	Load ¼	½	¾	Full
Single valve engines up to 400 kw	1.32	1.09	1.03	1.00
Four valve engines up to 400 kw	1.30	1.08	1.03	1.00
Unaflow engines up to 400 kw	1.10	1.02	0.980	1.00
Geared turbines up to 600 kw	1.57	1.25	1.05	1.00
Geared turbines 600 to 2000 kw	1.50	1.20	1.04	1.00

[a] These values are for indicated or brake loads and do not take into account variations in generator efficiencies under various load conditions. Generator efficiencies vary from 85 per cent at one quarter load to 95 per cent at full load.

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various capacities when operating under full load conditions. Table 2 gives the factors by which the actual water rates under full load operating conditions should be multiplied to obtain actual expected water rates under various load conditions. From the load curves and these tables it is a simple matter to

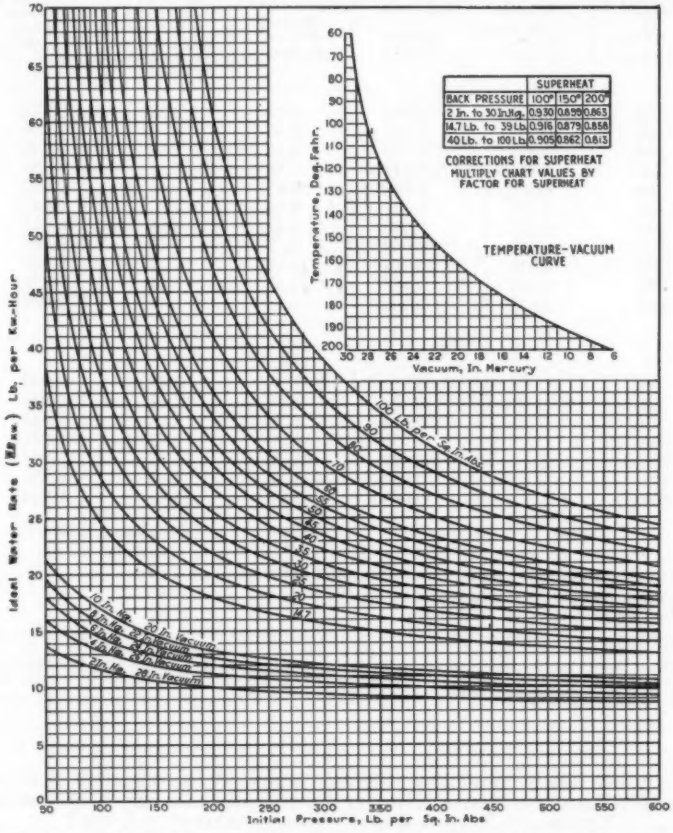
Ideal Water Rate (Wkw) Lb. per Kw-Hour
Initial Pressure, Lb. per Sq. In. Abs
Temperature, Deg. Fahr.
Vacuum, In. Mercury
SUPERHEAT
BACK PRESSURE 100° 150° 200°
2 In. to 30 In. Hg. 0.930 0.895 0.863
14.7 Lb. to 39 Lb. 0.916 0.879 0.858
40 Lb. to 100 Lb. 0.905 0.862 0.813
CORRECTIONS FOR SUPERHEAT
MULTIPLY CHART VALUES BY
FACTOR FOR SUPERHEAT
TEMPERATURE-VACUUM
CURVE
100 Lb. per Sq. In. Abs.
20 In. Vacuum
28 In. Vacuum

FIG. 5. IDEAL WATER RATE PER KILOWATT HOUR WITH DRY SATURATED STEAM FOR VARIOUS INITIAL PRESSURES AND BACK PRESSURES[a b]

[a] For ideal water rate per indicated horsepower hour, divide these values by 1.34.
[b] From Power from Process and Space Heating Steam, by L. A. Harding, A. S. H. V. E. TRANSACTIONS, Vol. 36, 1930.

develop curves showing the steam required by the electric generating or power units.

The next step is to determine the amount of heat available in this steam for heating, etc., after it has passed through the generating units. This can be done by deducting the heat transferred into work, plus that lost by radiation

from the heat in the steam entering the generating units. Since 2,546 Btu per hour = one horsepower hour and 3,415 Btu per hour = one kilowatt hour the heat extracted in performing work per pound of steam used may be obtained by dividing these figures by the water rates of the units per indicated horsepower hour or per kilowatt hour, respectively, of work done by the steam. From 1 per cent to 2 per cent may be deducted for radiation and other heat losses, depending upon the size of unit.

Example. A 200 kilowatt turbine is operating on saturated steam at 150 lb. gage pressure and consuming 40 lb. of steam per kilowatt hour delivered, at full load. Determine the heat available in the exhaust for heating.

Solution. The mechanical efficiency of the turbine is 0.935 and the efficiency of the generator is 0.934 (See Table 1), giving a combined efficiency of $0.935 \times 0.934 = 0.873$ which would mean $40 \times 0.873 = 34.92$ lb. per kilowatt hour of external work actually done by the steam. The total heat of the steam under the conditions as fed to the turbine is 1,195 Btu per pound.

$$\text{The heat extracted in external work} = \frac{3{,}415}{34.92} = 97.74 \text{ Btu per pound of steam.}$$

To this add 1½ per cent for heat lost by radiation, or $97.74 + 1.47 = 99.21$ Btu which is the total heat loss per pound of steam in passing through the turbine. This leaves $1{,}195 - 99.21 = 1{,}095.79$ Btu in each pound of the exhaust.

If the exhaust is to be used and condensed at atmospheric pressure there will be a further deduction of 180 Btu per pound for the heat of the liquid at this pressure. This leaves $1{,}095.79 - 180 = 915.79$ Btu per pound of steam fed to the turbine as being available in the exhaust for heating at atmospheric pressure.

In a similar manner the heat available per pound in the exhaust from an engine driven unit for heating may be estimated by dividing the heat equivalent per unit of work by the number of pounds of steam fed to the unit per unit of actual external work done, adding to this from 1 per cent to 2 per cent for radiation, and subtracting this sum from the total heat in a pound of the steam as fed to the unit and then subtracting from this remainder the heat per pound in the condensate or rejected steam leaving the heating system.

In the case of an engine-driven unit, the heat equivalent of a horsepower hour (2,546 Btu) should be divided by the water rate per indicated horsepower. If the water rate is given per kilowatt hour delivered by the generator, this may be reduced to pounds per indicated horsepower hour by dividing by 1.34 and multiplying by the mechanical efficiency of the unit and the efficiency of the generator.

Example. A 200-kw., engine-driven unit is operating under the same conditions as the turbine unit in the preceding example, and delivering one kilowatt hour on 37.8 lb. of water.

The mechanical efficiency of the unit is 0.905 and the efficiency of the generator is 0.934 (See Table 1). The water rate per indicated horsepower is therefore

$$\frac{37.8}{1.34} \times 0.905 \times 0.934 = \frac{37.8}{1.34} \times 0.845 = 23.84$$

The heat equivalent of the external work done per pound of steam is $\frac{2,546}{23.84} = 106.79$ Btu. To this add 1½ per cent for radiation, or $106.79 + 1.60 =$ 108.39 Btu which is the total heat lost per pound of steam in passing through the engine. Deduct this from the total heat per pound in the steam fed to the engine, or $1,195 - 108.39 = 1,086.61$ Btu.

DISCUSSION

C. W. OBERT (WRITTEN): It is fortunate that at this time when all possible economies are being sought in industry, Mr. West should have drawn attention to the much discussed question of heating with exhaust steam. With the trend of recent years toward utilization of central station electric power in public buildings, tremendous opportunities for steam power generation with heating as a by-product recovery are being lost, and the only advantages for the present prevailing practice are the lower initial cost of low-pressure power plant equipment, as compared with high-pressure power plant equipment, when the initial installation is made, and freedom from increased labor troubles from operation of the latter.

It is folly to think that small boiler plants need necessarily be a source of trouble or loss to the building management. If properly designed in the first place, the steam plant for lighting, heating, and producing refrigeration in the average hotel, hospital, store, factory or other public building will be found a source of comfort and economy. Competent labor for operation of such plants is obtainable for reasonable figures, although it is here pointed out that it does not pay to strive for the lowest cost of help; competent employees, paid a reasonable living wage, will be found a good investment in the operation of such small heating plants. Much of the criticism of small plants is caused by incompetent management.

It is evident that more study should be given to this phase of the heating problem. Under certain governing conditions, it is possible to combine to great economic advantage a high-pressure plant producing electric current as a by-product, with the purchase of public service electricity in the off-heating season. Careful analysis of the conditions should be made in each case by an engineer experienced in this class of work in order to ascertain what arrangement will work out to the best advantage. There are a great variety of operating plans and procedures to meet all classes of requirements, and every possible advantage should be taken of them. Mr. West's paper calls attention to an important study, and the engineering profession is indebted to him for his timely warning of a source of great national economic loss.

No. 898

SEMI-ANNUAL MEETING, 1931

THREE hundred members and guests responded to the call to meet in Swampscott, Mass., at the New Ocean House, June 22 to 25, when one of the finest technical programs ever prepared for a Society summer meeting was presented.

Special action which was taken by the members included acceptance of the Code for Testing and Rating Concealed Gravity Type Radiation (Steam Section); amendment of Regulations Governing the Committee on Research; and approval of additions to the Standard Test Code for Disc and Propeller Fans, Centrifugal Fans and Blowers.

During the three days all technical sessions were held from 9:30 a.m. until 12:30 p.m., and the papers presented covered a wide range of problems important to heating, ventilating and air conditioning engineers.

The Semi-Annual Meeting, 1931, was called to order by Pres. W. H. Carrier at the New Ocean House, on Tuesday morning, June 23, and D. S. Boyden, General Chairman of Arrangements, for the Massachusetts Chapter, expressed the wish of the local members that a most enjoyable time be had by everyone.

In the absence of John Howatt, Chairman of the Committee on Code for Testing and Rating Unit Ventilators, C. P. Bridges reported that a meeting of the Committee had been held to consider a tentative code but in the absence of unanimous agreement a definite report would not be given to the Society until a later date.

The report of the Committee on Rating and Testing Concealed Gravity Type Radiation was presented by Chairman R. N. Trane, who stated that the instructions given at the 37th Annual Meeting in Pittsburgh had been studied by the Committee and the Code revised so that it was now presented in two sections, a steam code and a tentative hot water code.

Each section being independent of the other, the steam section, he said, would be presented first and the changes from the original draft would be emphasized. In review Mr. Trane said that the Code provided for testing by the boiler load method and proposes rating heaters on the condensation basis. Since the original presentation was approved in principle, he stated, check tests have been made to determine the accuracy of the correction factor 1.3 and Professor Kratz would discuss this matter.

Prof. Kratz, with the aid of slides, presented the results of research on three questions which confronted the Committee after the Annual Meeting, 1931, relating to the construction of the test booth, testing in warm and cold walled rooms and the application of the 1.3 power rule. M. G. Steele stated that tests

which he had conducted corroborated the statement of Professor Kratz, regarding inlet temperatures of 60 to 90 F. J. F. McIntire inquired about the Committee's reason for having a humidity range of 25 to 60 per cent and thought that the 4 per cent variation shown in tests should be given as 4 per cent plus or minus. After an extended discussion by Messrs. Kratz, Trane and McIntire, L. A. Harding concurred in the belief that manufacturers would test all radiators in accordance with the Code and believed it unnecessary to specify a definite size of unit for test.

A discussion of no-load loss was participated in by Messrs. Steele, McIntire, Kratz and Trane. At the conclusion of the discussion the Committee was requested to take the suggestions offered for the steam heating section of the Code and submit the changes for discussion and approval at the Thursday morning session.

President Carrier then asked Mr. Trane to present the proposed Code for Hot Water. So numerous were the questions that it was decided to have the Committee do further work on the hot water code and present its report for adoption at a later meeting of the Society.

On Thursday morning President Carrier announced that R. N. Trane would report for the Committee on Code for Testing and Rating Concealed Gravity Type Radiation. Mr. Trane stated that the Committee held a meeting after the Wednesday morning session to discuss the suggestions which had been made and had included them in the report which he presented.

Vice-President F. B. Rowley took the chair, and additional proposals were made by Mr. Fleisher and Professor Smith, regarding the number of tests to be made and by J. F. McIntire regarding the test room to be used.

Professor Rowley then said that as the Code had been adopted in principle at the Pittsburgh meeting, the regular procedure would be followed upon receipt of the Committee's final draft, namely: have the Code printed and sent to the membership for approval by letter ballot vote.

By vote of the meeting this plan was approved unanimously.

President Carrier introduced W. H. Driscoll, Chairman of the Committee on Ventilation Standards, who stated that the Committee was making an informal report of progress and he would outline the work accomplished and the goal of the Committee. The object of the Committee's work he said was to prepare a Code on Ventilating Requirements for the use of legislative bodies and it is intended that the report will set up certain definite standards which legislators, engineers or members of the Society might use as a basis for or to assist in the preparation of bills for enactment that will conform to particular conditions in a given locality. Because of the requirements of different buildings, building construction, climates, etc., a uniform act for adoption of all legislatures would produce endless confusion, Mr. Driscoll pointed out. The standards to be set up, he said, must be sound, reasonable and logical and in order to safeguard the ventilation conditions, particularly in schools, it is the Committee's hope that its report in final form will be a bulwark against attack. The final report, he stated, will cover all classes of buildings to be ventilated and it is expected that the work of the Committee will result in giving the Society a comprehensive report for discussion at the Annual Meeting in January, 1932.

At the conclusion of the discussion, on motion of Mr. Myrick, properly seconded, it was voted that the Progress Report of the Committee be accepted and sent to all members and that the Committee be asked to carry on its work and render a final report for Society action at the earliest possible date.

Report of Committee on Research

The Chairman of the Committee on Research, C. V. Haynes, was introduced and briefly presented a report of the progress made since the Annual Meeting of the Society. He read a letter which was being sent to manufacturers interested in the Society's Research work and described a booklet being printed for the Committee on Research, giving a list of the projects under way, the history of the Laboratory and an index to papers published in the Society's TRANSACTIONS and JOURNALS which originated at the Research Laboratory or in co-operative institutions.

Mr. Haynes outlined the problems under investigation by the Committee on Research and announced that a new era in the Society's Research activities was beginning because of the action of the committee in appointing Prof. A. C. Willard as Technical Adviser.

A report prepared by F. C. Houghten, Director of the Research Laboratory, outlining the work at Pittsburgh and in cooperating institutions was presented.

Report of the A. S. H. V. E. Research Laboratory

By F. C. HOUGHTEN, *Director*

Since the 37th Annual Meeting of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, progress has been continued in connection with the twelve research projects which the Committee on Research has under consideration. Active work has been carried on in the Laboratory at Pittsburgh or in the co-operating university laboratories on most of the projects. Those not actively under investigation in the Laboratory have been given consideration by the Technical Advisory Committees with a view to developing plans for the future.

The work has resulted in the publication, or release, of ten technical papers.

1. **Heat Transmission.**—(Heat Received by and Emitted by Buildings in Relation to Living Comfort)—*Technical Advisory Committee:* L. A. Harding, *Chairman;* R. E. Backstrom, F. B. Rowley, A. E. Stacey, J. H. Walker.

a. Determination of Heat Transfer through Walls by the Guarded Hot Box Method.

This work was continued at the University of Minnesota under the direction of F. B. Rowley, in co-operation with the Research Laboratory of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS. A number of concrete and stone masonry walls were tested, giving additional data for publication.

b. Film Conductance Coefficients for Various Types of Surfaces with Still and Moving Air.

This work was continued at the University of Minnesota in co-operation with the Laboratory and additional data have been collected giving the effect of angles of wind impingement on the film conductance coefficient. A paper on this phase of the work was recently released for publication.

c. Effect of Aging on Conductivity of Concrete.

Additional tests were made in the Laboratory at Pittsburgh to determine the variation in conductivity of concrete with time. A paper dealing with this variation, and discussing other factors affecting the conductivity of concrete, has been released

for publication. This paper gives a review of conductivity values published by various investigators.

d. Effect of Heat Capacity of a Structure on Heat Flow into a Building from Solar Radiation.

Since the annual meeting, the Laboratory has worked on the analysis of the available data on this subject. A comprehensive plan for additional work to be carried on during the summer months has been outlined by the Technical Advisory Committee.

2. Ventilation of Garage and Bus Terminals.—*Technical Advisory Committee:* E. K. Campbell, *Chairman;* D. S. Boyden, H. P. Gant, W. T. Jones, W. C. Randall.

A study of garage ventilation was undertaken last year at Washington University, St. Louis, in co-operation with the Research Laboratory of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS. The object of the study was to develop the relations between the quantity of air handled by the ventilating system, its temperature and velocity, the location of intakes and exhausts, the quantity of outside air handled and the concentration and distribution of carbon monoxide and gasoline vapors in the garage. An effort is being made to develop a co-operative study of the subject with some institution located in or near a large city in a northern climate. It is hoped that plans for this work will be perfected in time to begin the study early in the next heating season.

3. Pipe and Tubing (Sizes) Carrying Low Pressure Steam or Hot Water.—*Technical Advisory Committee:* S. R. Lewis, *Chairman;* J. C. Fitts, F. E. Giesecke, H. M. Hart, C. A. Hill, A. P. Kratz, W. K. Simpson.

a. Capacity of Pipe for Various Parts of a Steam Heating System.

The Laboratory recently completed a study which gave the relation between pressure drop and condensation for various rates of handling air in return risers and dry return mains. These data are now being further analyzed with a view to revising Pipe Size Tables in the A. S. H. V. E. GUIDE so as to conform with the latest findings of the Laboratory. Additional work is being done at the Laboratory in Pittsburgh using copper pipe and fittings to establish flow relationships for this type of pipe and fittings in steam heating systems.

b. Capacity of Pipe for Various Parts of a Hot Water Heating System.

This work is being carried on by F. E. Giesecke and his associates at the Agricultural and Mechanical College of Texas in co-operation with the Laboratory. The object of the study is to develop usable pipe size tables for the proper design of all parts of hot water heating systems. A paper, Sizing Pipes and Orifices for Gravity Hot Water Heating Systems, by Elmer G. Smith, gives usable tables for laying out hot water heating systems and orifice sizes for distributing the flow of water to radiators of various sizes in different parts of the system.

A study is also being made at the Agricultural and Mechanical College of Texas of the frictional resistance to flow of water through the various types of iron, brass and copper fittings on the market. A paper, Friction Heads in One-Inch Standard Cast-Iron Tees, by F. E. Giesecke and W. H. Badgett, gives the frictional resistance for divided flow through cast-iron tees. This work is being continued on other types of tees, including copper, and should give results which have long been needed.

c. Heat Emission from Bare and Covered Copper, Brass and Iron Pipe.

With the increased use of copper and brass pipe in heating systems, there is a growing need for data concerning heat loss from such pipe either bare or covered. Since the external surface area of the copper tubing used is smaller than the same size iron pipe, and since it has a lower emissivity, the heat loss will be less for uncovered copper than for uncovered iron pipe. Determination of these losses is being made in the Laboratory at Pittsburgh.

4. Atmospheric Dust and Air Cleaning Devices.—*Technical Advisory Committee:* H. C. Murphy, *Chairman;* Albert Buenger, Philip Drinker, Dr. E. V. Hill, H. B. Meller, Dr. S. W. Wynne.

Dean A. S. Langsdorf at Washington University, in co-operation with the Research Laboratory, is undertaking the development of a more sensitive and accurate method

of determining air dustiness. This work has been under way during the past year and gives promise of very valuable developments.

The Technical Advisory Committee on this subject has been actively engaged in developing a program for future activity. Its plans include the development of a code for testing air-cleaning devices, but since no instrument now on the market is entirely satisfactory for all purposes, it is proposed by the committee to cover by specifications the design of an instrument of a jet-impingement type for industrial and field work, and of an instrument of the filter type for investigation of atmospheric pollution. The committee hopes to correlate the various interests and activities in the field of atmospheric dust and smoke study.

5. Radiation.—(Direct and Indirect).—*Technical Advisory Committee:* John Holton, *Chairman;* R. M. Conner, R. E. Daley, R. V. Frost, F. E. Giesecke, H. F. Hutzel, A. P. Kratz, J. F. McIntire, W. T. Miller, R. N. Trane.

The Research Laboratory recently entered into a co-operative agreement whereby the Laboratory's research work dealing with the heating effects of direct and indirect radiation will be studied at the University of Illinois. This arrangement makes available the excellent facilities for studying radiation developed at the University of Illinois during the past several years.

A study of the heating effect of radiators will be made in special test rooms. This includes a continuation of the study in the room-heating testing plant of the performance characteristics of the various types of cast-iron radiators and also of various types of non-ferrous radiators having heating elements made of copper, brass, and aluminum assembled in enclosures acting as convectors.

These tests are conducted under heating service conditions in actual rooms with outside air temperatures of approximately zero degrees Fahrenheit and with inside air temperatures taken at the 30-in. level, of approximately 68 F. Comparisons between the different radiators tested are made on the basis of weight of steam condensed per hour and the air temperature gradient between ceiling and floor levels. It should be noted especially that, while the air temperature at the 5-ft or breathing-line level is always recorded, it is the air temperature at the 30-in. level which is kept the same, at 68 F, and is used as the basis for comparison.

Professor Willard is also proceeding with the erection and operation of a test booth in accordance with the specifications contained in the Code for Testing and Rating Concealed Gravity Type Radiation (Edition of June, 1931). In this test booth the same radiators which have been tested in the room heating testing plant, will be again tested in accordance with the proposed code for the purpose of obtaining correlation factors between the two plants with various types of radiators.

6. Infiltration in Buildings.—*Technical Advisory Committee:* G. L. Larson, *Chairman;* J. E. Emswiler, F. E. Giesecke, W. C. Randall, W. A. Rowe, J. G. Shodron, Ernest Szekely, M. S. Wunderlich.

The Research Laboratory's program for testing various types of walls for air leakage, as outlined several years ago and studied in the Laboratory at Pittsburgh and by Prof. G. L. Larson at the University of Wisconsin in co-operation with the Laboratory, was completed last year. The work at Wisconsin has been aimed toward the clearing up of certain phases of air leakage around weather-stripped and non-weather-stripped windows. The paper Air Infiltration through Double-Hung Wood Windows, by G. L. Larson, D. W. Nelson and R. W. Kubasta, includes the results of this investigation.

Professor Larson is also making a study of the airation of buildings with special reference to school-type buildings and open window ventilation. Readings are taken to determine the variation in the amount of air entering or leaving through an open window at various wind velocities. For this study, two classrooms of about the same size have been selected, one on the north side and one on the south side of the second floor of a four-story building which has a fully exposed and heated basement having outside openings. It is assumed that because the rooms are near the center of the height of the building the *chimney effect* is as small as it would be at any other place in the building. The inlet ducts from the ventilating system are kept closed; the outlet ducts so far have been kept open. One window in each room is

opened a definite height as used for open window ventilation. An anemometer is set in each opening and readings are taken at five minute intervals. At the end of each half hour, anemometers are interchanged so as to eliminate any difference in the instruments.

7. **Measurement of Air Flow Through Registers and Grilles.**—*Technical Advisory Committee:* John Howatt, *Chairman;* John Aeberly, C. A. Booth, L. E. Davies, D. E. French, J. J. Haines.

Prof. L. E. Davies at Armour Institute of Technology, Chicago, in co-operation with the Research Laboratory is continuing the study of air flow through registers and grilles. Since the 37th Annual Meeting of the Society this study has been directed toward the development of a satisfactory method of measuring the air delivery of a unit ventilator. In connection with the study of air delivery from unit ventilators and the study of air delivery through grilles into a room, further studies have been made of the application of the formula for measuring air delivery through grilles presented in the paper, Measurement of the Flow of Air through Registers and Grilles, by L. E. Davies. (See A.S.H.V.E. TRANSACTIONS, Vol. 36, 1930.)

In most cases the formula has been found to apply satisfactorily, but in the following few cases it does not seem adequate:

1. Where the grille design is characterized by largeness of details
2. Where it is difficult or impossible to determine accurately the amount of free area, as in ornate designs or complex structures such as unit heaters or ventilators;
3. Where the nature of the approach is such as to cause a very erratic velocity distribution across the face of the grille.

A study has been made of the application of a cone, placed over the grille work, into which the anemometer is fitted to measure the air delivery from such grilles as in the foregoing cases. The use of such a cone offers a possibility of success. A paper, The Measurement of the Flow of Air Through Registers and Grilles (*Part II*), by L. E. Davies, covers additional phases of the work carried on at Armour Institute.

8. **Air Conditions and Their Relation to Living Comfort.**—*Technical Advisory Committee:* C. P. Yaglou, *Chairman;* O. W. Armspach, W. L. Fleisher, Dr. E. V. Hill, Dr. R. R. Sayers.

The work on this study is divided into two distinct phases; *first,* the study of vital characteristics of the atmosphere by C. P. Yaglou and Philip Drinker at the Harvard School of Public Health, in co-operation with the Research Laboratory; and *second,* a study of the relation between atmospheric conditions and thermal exchanges between the human body and its atmospheric environment made in the Laboratory at Pittsburgh.

It is also hoped that, in connection with this study, daily observations may be made of the atmospheric ionization indoors and outdoors at two other localities, preferably Pittsburgh and Portland, Oregon, in addition to the observations which will be continued in Boston. These data are to be correlated with incidence of illness and death rates in the localities from seasonal diseases in the various localities.

The study of the relation between atmospheric conditions and thermal exchanges between the human body and its atmospheric environment is being continued at Pittsburgh. A paper, Thermal Exchanges between the Bodies of Men Working and the Atmospheric Environment, by F. C. Houghten, W. W. Teague, W. E. Miller and W. P. Yant, resulting from this work, appeared in the March issue of the *American Journal of Hygiene.* Another paper, Heat and Moisture Losses from Men at Work and Application to Air Conditioning Problems, appears in the TRANSACTIONS OF THE SOCIETY, Vol. 37, 1931.

The study of thermal exchanges between the bodies of children of school age and the atmospheric environment, and the application of the effective temperature scale and comfort zone to children, is now nearly completed, and a paper on this subject will be ready for publication some time during the next few months. Additional plans for this investigation include a study of physiological reactions of persons to low temperatures (within and below the comfort zone), the effect of moisture content of the atmosphere on humans (within the range of 40 to 70 per cent relative humidity); and the effect of cold walls on the comfort zone for persons at rest and normally clothed.

9. Oil and Gas Burning Devices.—*Technical Advisory Committee:* L. E. Seeley, *Chairman;* R. M. Conner, P. E. Fansler, R. S. Franklin, R. V. Frost, R. C. Morgan, H. F. Tapp.

This study is carried on by Prof. L. E. Seeley of Yale University in co-operation with the Research Laboratory of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, and the *American Oil Burner Association.* The investigation has for its purpose the development of a standard code for testing and possibly rating oil burning boilers and furnaces, and the development of additional facts concerning combustion space, atomization, and air mixture, and their relation to efficient operation.

As a result of this study, a code was presented at the 37th Annual Meeting of the Society for testing low pressure steam heating boilers burning oil fuel. A paper, Study of Performance Characteristics of Oil Burners and Low Pressure Heating Boilers, by L. E. Seeley and E. J. Tavanlar, is published in the TRANSACTIONS OF THE SOCIETY, Volume 37, 1931. The scope of the work in this study is being broadened and will be continued.

10. Sound in Relation to Heating and Ventilation.—*Technical Advisory Committee:* F. B. Rowley, *Chairman;* Carl Ashley, Warren Ewald, R. F. Norris, G. T. Stanton.

This is a new project under consideration by the Research Committee, and no definite plan of attack has yet been developed. The study has for its purpose the development of facts concerning noises resulting from heating and ventilating devices and methods of detecting, measuring and locating the source of them.

11. Correlating Thermal Research.—*Technical Advisory Committee:* R. M. Conner, *Chairman;* D. S. Boyden, P. D. Close, J. C. Fitts, W. T. Jones, H. T. Richardson, Perry West.

This is a new committee appointed to receive and administer the work of cataloging and maintaining a file and bureau of information concerning available published data and research activities. This work is now carried on by the *Association for Correlating Thermal Research,* which association will be disorganized after transferring the work to the Research Laboratory and providing for this year's budget.

Report of Technical Advisory Committee on Radiation

JOHN H. HOLTON, Chairman

Prior to the Summer Meeting, 1931, a letter was sent to all members of this committee outlining the problems to be discussed. Combining replies received from these letters, and personal interviews with members who attended this committee meeting, the following recommendations represent the majority opinion.

RECOMMENDED:

1.—That for the present, no action be taken on the commercial testing of radiation, either exposed or concealed, by the A. S. H. V. E. Laboratory. Our laboratory facilities can be used to better advantage along lines tending to clarify approved methods of testing all forms of radiation.

2.—That codes for testing all forms of so-called "radiation" both exposed and concealed, be developed and approved by the Society. With these as a basis, other agencies may then be in a position to do all the necessary testing, then if desired, the A. S. H. V. E. may consider the proper organization and equipment to undertake this commercial testing. Until such codes are a part of the approved program of the Society, we are not in a position to render needed service to the industry.

3.—That in view of the divided opinion and the many unknown factors involved in the effectiveness of the various heating mediums, it is suggested, at least for the present, we go on record as favoring a condensation method properly controlled as the measurement of the total heat delivery from such equipment. In other words, a condensation rating represents a load method and measures the amount of heat required by, and delivered to the heating equipment.

4.—That the Society continue to authorize the study of the utilization or effectiveness

of the heat delivered by various types of heating equipment. This involves further investigations of the so-called heating effect, and should determine:

(a) Definition of heating effect.
(b) Method of measurement.
(c) Correct location of temperature control point.
(d) Effect of variation in wall construction and wall temperatures. This work must lead to a standard procedure or code for the testing of heating effect as defined.

5.—That in view of the foregoing, a Committee be appointed to revise and bring up to date in the form of a code, the present exposed radiator test method, having in mind the desirability of correlating this code with that now under consideration on concealed radiation. We further recommend that these test codes be published in separate pamphlets available to all members for insertion in their book of standard codes.

6.—That definite tests be authorized to prove whether or not the so-called hot room method of testing gives the correct results, bearing in mind that if possible, this method should be approved due to its simplicity and low cost. Such a method would be within reach of all manufacturers, while the high cost of the cold room method would tend to retard its universal acceptance. Such work is, at present, under way in the University of Illinois, and should be continued on both concealed and exposed radiation units.

While this Committee could make further recommendations for additional program at this time, it feels advisable to await the outcome of the development of these codes and test methods before definitely recommending further tests. The results of the above will clearly indicate the trend of our future research work.

Report of Technical Advisory Committee on Air Conditions in Relation to Comfort and Health

C. P. YAGLOU, Chairman

During the past two years the Harvard School of Public Health in cooperation with the Society's Research Laboratory has been studying the problem of ionization in relation to air conditioning and health. Several phases of the work dealing with diurnal and seasonal variations in the ionic content of outdoor and indoor air, ionic changes due to respiratory and metabolic processes, and the de-ionizing effects of air conditioning methods, have already been completed. A paper on these is being prepared for the Annual Meeting, 1932, in January, and it is scheduled for publication in the September issue of the JOURNAL of the Society.

The phases of the work now being studied are:

a. Physiologic, bacteriologic and bio-chemical effects of artificially ionized and de-ionized air.
b. Relationship between seasonal variation in atmospheric ionization and seasonal variation in the incidence of respiratory diseases.
c. Methods for artificial ionization of air.

Other plans of the committee for future work include the physiologic, bacteriologic and bio-chemical effect of cold and of humidity on human beings and animals.

Very little information is available on the effects of cold upon the organism. The region covered by the Research Laboratory extends from 80 deg E. T. upward. In the opinion of the committee, the Research Laboratory should complete this work, which, in conjunction with the study on ionization, may throw some light on the etiology of respiratory diseases.

Concerning the influence of humidity on the organism, the committee proposes a thorough study of the effects of changes in humidity through a range of 60 per cent at three different temperatures, namely 40 F, 70 F and 95 F, with the possibility of determining the optimum humidity zone. Observations are to be made on men as well as on animals. The observations on animals are to include the course and resistance to pneumococcus and enteric infections at high and low humidities, growth, reproduction, etc.

Perry West presented the following resolution:

"Whereas, many of the problems concerning the value and worth of ventilation and air conditioning from a comfort standpoint have been solved by our Research Laboratory, and *Whereas,* there remains to be solved the very important problems concerning the value and worth of ventilation and air conditioning from the health, efficiency and longevity basis; *Therefore be is Resolved,* that it is the sense of this meeting that the Society give serious consideration to the establishment of the worth of ventilation and air conditioning from a health and efficiency basis and that the Research Committee be requested to study and report on that matter at the next Annual Meeting."

This resolution was seconded and when voted upon, unanimously approved.

W. H. Driscoll assumed the chair and Prof. F. B. Rowley outlined the changes in the regulations governing the Committee on Research which had been approved by the Society in January, 1930. The Committee desired the Society to vote upon the following suggestions:

Section I—7, which reads as follows:

Quorum

Seven members of the Research Committee shall constitute a quorum.

To be amended to read:

Quorum

Five members of the Research Committee shall constitute a quorum.

Eliminate from Section II, 6,

All papers, findings or reports resulting from the work of the Research Committee shall when published by the Society be headed by a picture of the Laboratory at Pittsburgh, Pa.

To Section I, 1, add

(f) The outgoing Chairman if serving in that capacity during the last year of his three (3) year term on the Committee shall without election become an additional member of The Committee on Research for one (1) year.

It was regularly moved and seconded that the changes requested by the Committee on Research be approved by the Society.

Under the heading of new business a special committee appointed by the Council to consider the resolution of the *National Association of Fan Manufacturers* with respect to amending the Standard Test Code for Disc and Propeller Fans, reported that it had studied the proposal and on motion of L. A. Harding, seconded by W. T. Jones, it was voted that the following addition be made to the Standard Test Code:

Following the paragraph *Exhausting Fans* insert new paragraph—*Centrifugal Fans with Inlet Boxes:*

Centrifugal fans with inlet boxes shall be tested with inlet boxes in place. A duct (or ducts for double inlet fans) of the same size and form as the inlet box opening shall be connected thereto, as shown on Plate *G*. The inlet test duct shall be not less in length than the mean of its two sides. The open end shall be provided with a nozzle inlet formed of cylindrical sides curved to a radius of not less than one-half of the short dimension of the duct, and not less than a 75 deg arc. Readings shall be taken in said test duct at a point one-half of a mean side from the open end exclusive of the length of the nozzle. The readings are to be taken in accordance with

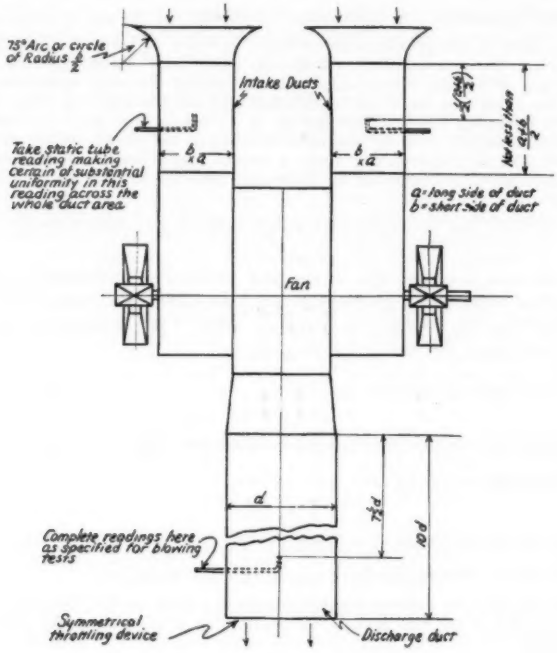
75° Arc or circle of Radius d/2
Intake Ducts
Take static tube reading making certain of substantial uniformity in this reading across the whole duct area.
b/4a
b/4a
a = long side of duct
b = short side of duct
Fan
d
2½d
10d
Complete readings here as specified for blowing tests
Symmetrical throttling device
Discharge duct

PLATE G

Plate *G* and the calculations made in accordance with the calculation sheet for Centrifugal Fans Exhausting. (See Plates *A*, *B* and *C*.)

Ratings based on such tests shall be clearly stated as applying to *Centrifugal Fans with Inlet Boxes* and include the loss in the boxes.

Paragraph *Exhausting Fans,* ninth line—place an asterisk after the word *adding* with the following footnote to which it refers:

This addition is algebraic if, instead of calculated absolute pressures, observed readings are used, those below atmospheric pressure being negative, those above atmospheric pressure positive, and the velocity head always positive.

Paragraph *Disc and Propeller Fan Tests* under the sub-heading *Test Set-up,* fourth paragraph in last sentence, omit the word *static.* The revised form of this sentence to be—

The pressure in the large chamber shall be determined by the reading of an impact tube. (See Plate *B*.)

Resolutions

Mr. Harding offered the following resolutions which were seconded by H. M. Nobis and unanimously carried:

RESOLVED, That the thanks of the members attending the 37th Semi-Annual Meeting of the Society be officially expressed to the Massachusetts Chapter for the hospitality extended to the Society on this occasion and especially to the local Chapter Committees for the excellent entertainment and program provided; the fine service rendered by the hotel management; the co-operation of the authors of the papers presented and the assistance of the press in giving fine publicity to the proceedings of the meeting and the activities of the Society.

As no further business was offered, President Carrier declared the 37th Semi-Annual Meeting adjourned.

PROGRAM SEMI-ANNUAL MEETING

AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS

NEW OCEAN HOUSE, SWAMPSCOTT, MASS., JUNE 22-25, 1931

TECHNICAL

(Daylight Saving Time)

Monday, June 22

9:30 A.M. Registration.
12:30 P.M. Luncheon (Officers, Council and Authors).
2:00 P.M. Council Meeting.
3:00 P.M. Meeting Committee on Research.

Tuesday, June 23

8:30 A.M. Registration.
9:30 A.M. Technical Session.
Greeting by D. S. Boyden, *Chairman*, Committee on Arrangements.
Response by President W. H. Carrier.
Flow of Steam through Orifices into Radiators, by S. S. Sanford and C. B. Sprenger.
Friction Heads in One-Inch Standard Cast-Iron Tees, by F. E. Giesecke and W. H. Badgett.
Sizing Pipes and Orifices for Gravity Hot Water Heating Systems, by E. G. Smith.
Using Gas-Vapor Mixtures for Heating Purposes, by C. A. Dunham.
Report of Committee on Increase of Membership, C. W. Farrar, *Chairman*.
Report of Committee on Testing and Rating Unit Ventilators, John Howatt, *Chairman*.
7:00 P.M. Conference of Chapter Representatives.
7:00 P.M. Meeting of Committee on Ventilation Standards.

Wednesday, June 24

8:00 A.M. Meeting of Nominating Committee.
9:30 A.M. Technical Session.
Essential Elements for Determining Heating Plant Requirements, by F. B. Rowley.
Report of Committee on Rating and Testing Concealed Gravity Type Radiation, R. N. Trane, *Chairman*.
Steam Condensation an Inverse Index of Heating Effect, by A. P. Kratz and M. K. Fahnestock.
Experiments to Determine Heating Effect Factor, by R. N. Trane and C. J. Scanlan.
Study of Performance Characteristics of Oil Burners and Low Pressure Heating Boilers, by L. E. Seeley and E. J. Tavanlar.

Thursday, June 25

9:30 A.M. Technical Session.
Report of Committee on Ventilation Standards, by W. H. Driscoll, *Chairman*.
Report of Committee on Research, C. V. Haynes, *Chairman*.

XUM

Reports of Technical Advisory Committees.
Heat and Moisture Losses from Men at Work and Application to Air Conditioning Problems, by F. C. Houghten, W. W. Teague, W. E. Miller and W. P. Yant.
Infiltration through Double Hung Wood Windows, by G. L. Larson, D. W. Nelson and R. W. Kubasta.
Predetermining the Airation of Industrial Buildings, by W. C. Randall and E. W. Conover.
Measurement of the Flow of Air through Registers and Grilles, by L. E. Davies.
Unfinished Business.
New Business.
Adjournment.

ENTERTAINMENT

Monday, June 22

12:30 P.M. Luncheon (Officers, Council and Authors).
8:30 P.M. Informal Reception and Dance.

Tuesday, June 23

10:00 A.M. Clock Golf.
1:30 P.M. Coach and Boat Trip to Marblehead and Gloucester, including the famous North Shore Drive.
2:00 P.M. Golf—Research Cup Tournament.
8:00 P.M. Bridge Party for Ladies.

Wednesday, June 24

9:00 A.M. Motor Trip to Historical Boston and Boston Navy Yard to inspect the frigate *Constitution* or *Old Ironsides.*
2:00 P.M. Golf—Kickers' Handicap.
7:00 P.M. Banquet and Dance at New Ocean House (informal).

Thursday, June 25

9:00 A.M. Special Golf Tournament—Ladies.
2:00 P.M. Trip to historic Lexington, Concord and Salem.

Committees

General Committee for Semi-Annual Meeting 1931

D. S. Boyden, *Chairman*
W. T. Jones, *Vice-Chairman*

Reception and Registration: J. S. Webb, *Chairman;* E. Q. Cole, J. A. Reardon, W. B. Leland, Alfred Kellogg, C. R. Swaney.

Finance: W. T. Jones, *Chairman.*

Ladies and Hostesses: Mrs. E. A. Dusossoit, *Chairman;* Mrs. Leslie Clough, Mrs. W. T. Jones, Mrs. J. W. Brinton, Mrs. T. F. McCoy.

Sports: E. A. Dusossoit, *Chairman;* F. R. Ellis.

Entertainment: J. W. Brinton, *Chairman;* H. B. Wiegner, W. N. Murray, C. T. Flint.

Publicity: Leslie Clough, *Chairman;* R. F. Gifford, A. T. Papineau.

Golf: J. F. Tuttle, *Chairman;* A. C. Bartlett, R. S. Franklin, A. A. Klonower, W. R. Illig.

Transportation: T. F. McCoy, *Chairman;* P. C. Pocock, E. W. Berchtold.

No. 899

A.S.H.V.E. STANDARD CODE FOR TESTING AND RATING CONCEALED GRAVITY TYPE RADIATION (STEAM CODE)

COMMITTEE: R. N. Trane, *Chairman*, E. H. Beling, W. F. Goodnow, John Holton, Hugo Hutzel, A. P. Kratz, E. J. Vermere, O. G. Wendel

A. PURPOSE

1. **Object of Code.** The object of this code is to provide a practical method for testing and rating concealed radiators which may readily be used by manufacturers and which will provide the necessary information for the heating engineer in the design of the heating plant. This code describes a boiler load method of rating.

2. **Definition.** The term concealed radiator[1] as used in this code shall apply to any type of radiator installed within a wall, casing or enclosure having an inlet and outlet for the circulation by gravity of the air in the room to be heated.

B. APPARATUS

3. **The Purpose of the Test Booth** shall be solely to maintain constant and uniform conditions about the radiator.

4. **Description of Test Booth.**[2] The test booth shall be 12 ft x 15 ft in floor area, and shall have a ceiling height of 9 ft. A tolerance of ± 10 per cent in any dimension is permissible. The floor of the booth shall be set up in a larger room (see Paragraph 5), at least 1 ft above the room floor. One side of the booth (either a 12-ft or a 15-ft side) shall be open with a shield projecting down from the top at least one foot. The walls and ceiling shall have a smooth, close surface, and shall be painted with oil paint. The air in this test booth shall be free from draft.

5. **Room Enclosing Test Booth.** The test booth shall be set up in a larger room and in such a position as not to be exposed to the direct rays of the sun. The distance between any test booth wall and the wall of the surrounding room shall not be less than 2 ft and the ceiling of the test booth shall not be less than 1 ft from the ceiling of the larger room. The air temperature in this larger room shall be taken at the three sides of the test booth at a level of 5 ft from the floor of the larger booth at a distance of 12 in. from the test booth walls and shall not show a variation to exceed ± 3 deg (F) during the course of a test.

6. **Location of Radiator in Booth.** The concealed radiator shall be set up in accordance with the manufacturer's specifications, adjacent to the wall opposite the open side of the test booth.

7. **Radiator Casing.** The radiator shall be placed either in a casing with grilles as furnished by the manufacturer, or if a casing is not furnished, it shall be placed in a box made of material simulating the catalog specifications.

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[1] The terms *radiation* and *radiator* are used in the commercial sense and not in the scientific sense in this code.

[2] A refrigerated testing room with walls having a construction typical of walls used in an actual building may be used instead of the warm wall test booth.

Report of Committee presented and adopted at the Semi-Annual Meeting of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, Swampscott, Mass., June, 1931.

C. GENERAL TEST PROCEDURE

8. **Relative Humidity.** The relative humidity at the time of test shall be within the limits of 25 per cent and 60 per cent, and the air motion shall be limited to natural circulation.

9. **Air Temperature.** The heater shall be tested with an inlet air temperature of not less than 60 F nor more than 80 F. These temperatures shall be measured with thermometers placed 12 in. apart (minimum of three) throughout the length of the inlet opening midway between top and bottom, and 3 in. in front of the inlet.

10. **Correction for Inlet Air Temperature.** When the inlet temperature differs from 65 F, the capacity of the radiator under the test condition shall be converted to the capacity under standard conditions as explained in Paragraph 29.

11. **Thermometers.** All thermometers used during tests shall be calibrated by comparison with a standard thermometer. They shall be graduated with divisions of not more than 1 deg.

12. **No-Load Test.** After each test a no-load test shall be run to determine the condensation in the piping alone which must be subtracted from the gross condensation from the radiator in order to obtain the net condensation.

D. PROCEDURE FOR TESTING

13. **Piping Diagram.** Fig. 1 shows the connections and equipment prescribed for supplying steam to the radiator and measuring the condensate. All instruments shown shall be used and shall be installed in the relative position indicated.

14. **The Supply Piping** from separator to radiator and return piping from radiator to receiver shall be as short as possible (not exceeding 2 ft in length).

15. **The Return Piping** from radiator shall pitch downward toward receiver to assure proper draining of condensate. There shall be no intervening valve in the return line between the radiator and receiver.

16. **The Receiver.** The receiver shall be not greater than 2 in. in diameter and not longer than 14 in. It shall be fitted with a gage glass of outside diameter not exceeding ⅝ in.

17. **Insulation.** All piping from the separator to the radiator and from the radiator to and including the receiver shall be insulated with 2 in. of hair felt or equivalent.

18. **The Separator** shall be of liberal capacity. The steam throttle or low pressure reducing valve shall be of a type suitable for close control. A manometer shall be used on the radiator side of the throttling valve.

19. **Steam Temperature.** The temperature of the steam shall be measured by a thermometer with the bulb in direct contact with the flow of steam. Thermometers shall be within 6 in. of the radiator inlet and outlet.

20. **Steam** shall be supplied from a source of sufficient capacity to prevent sudden changes in pressure.

21. **Steam Pressure.** Radiators shall be tested with steam at 15.7 lb absolute pressure. In order to introduce the required superheat, there shall be not less than 5 lb pressure drop through the throttle valve. If the quality of the steam supplied is so low that the required superheat is not introduced by the 5 lb throttling, the line pressure shall be raised until the required superheat is introduced. The superheat shall be not less than 2 deg nor more than 5 deg and be obtained by an approved method such as wire drawing through a controlled valve from a higher pressure (as shown in Fig. 1).

22. **Condensate.** The condensate level indicated by the gage glass shall be at the level established at the start of the test at the time of obtaining each periodic condensate weight. At no time during the test shall the water level of the receiver vary more than 2 in. from the level established at the start of the test. The condensate shall pass to the weighing vessel with provision made that no condensate shall be lost by evaporation. The weight of condensate shall be read to 0.01 lb.

23. **Air Venting.** The radiator shall be vented continuously by means of an air cock through the regular vent opening provided, or through a cock at the top of the condensate receiver. The vent shall discharge outside the radiator enclosure.

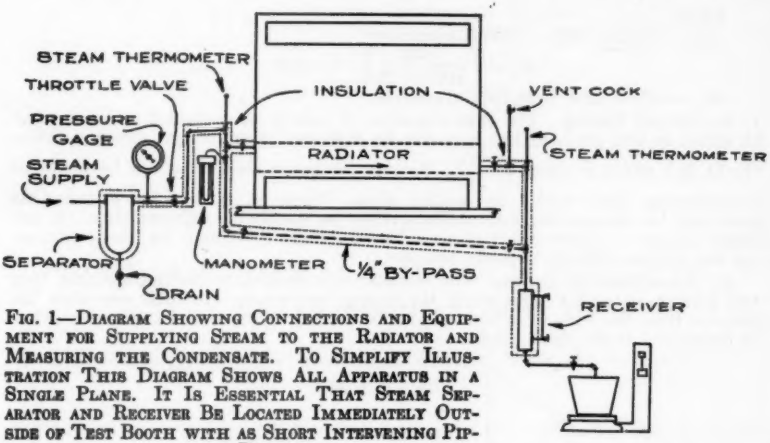

FIG. 1—DIAGRAM SHOWING CONNECTIONS AND EQUIPMENT FOR SUPPLYING STEAM TO THE RADIATOR AND MEASURING THE CONDENSATE. TO SIMPLIFY ILLUSTRATION THIS DIAGRAM SHOWS ALL APPARATUS IN A SINGLE PLANE. IT IS ESSENTIAL THAT STEAM SEPARATOR AND RECEIVER BE LOCATED IMMEDIATELY OUTSIDE OF TEST BOOTH WITH AS SHORT INTERVENING PIPING AS POSSIBLE

24. Start of Test. The test shall begin after the room temperature and condensation rate have remained constant for a period of 30 min.

25. The Separator shall be drained at 5-min intervals.

26. Duration of Test. A test shall be of not less than one hour duration during which each successive periodic condensate weight shall be within the limits of ± 5 per cent of the average of the periodic condensate weights. The condensate shall be weighed each 10 min during a test.

27. Output Under Test Conditions. The heating capacity of a concealed radiator under test conditions shall be determined by the following formula:

$$H_{ts} = W_s h_{fs} \tag{1}$$

where

H_{ts} = heat emission for steam under test conditions, Btu per hour
W_s = weight of condensate, pounds per hour
h_{fs} = latent heat of evaporation of steam corresponding to absolute pressure in radiator, Btu per pound

28. Correction Factor. The correction factor (C_s) for converting the output at test conditions to the output at standard conditions shall be determined by the following formula:

$$C_s = \left[\frac{215.3 - 65}{t_s - t_i}\right]^{1.3} = \left[\frac{150.3}{t_s - t_i}\right]^{1.3*} \tag{2}$$

where

C_s = correction factor
t_s = saturated steam temperature as of test, degrees Fahrenheit
t_i = average inlet air temperature as of test, degrees Fahrenheit

29. Output Under Standard Conditions. The output under standard conditions (H_s) shall be determined as follows:

$$H_s = C_s H_{ts} \tag{3}$$

30. Example. The following example will illustrate the use of the equations in Paragraphs 27, 28 and 29, inclusive:

Let

W_s = 10.03 lb
t_i = 70.0 F
t_s = 215.0 F
h_{fs} = 970 Btu per pound (at 215.0 F)

* Although the exponent 1.3 applies specifically to direct radiators, it also applies with reasonable accuracy to convectors.

Then

$$H_{ts} = 10.03 \times 970. = 9729 \text{ Btu per hour}$$

$$C_s = \left[\frac{150.3}{215.0 - 70}\right]^{1.3} = 1.048$$

$$H_s = 9729 \times 1.048 = 10{,}193 \text{ Btu per hour}$$

31. **Output Rating.** The heat emission or output of concealed radiators shall be stated in Btu per hour but may also be expressed in equivalent direct radiation (E. D. R.) which is equal to $\frac{H_s}{240}$. Ratings for larger sizes must not be made by extrapolating from results on smaller sizes. However, ratings for intermediate sizes may be obtained by interpolation from the results on adjacent sizes. A sufficient number of sizes should be tested to definitely determine the curve expressing the relation between capacity and size.

32. **Description of Grilles.** The manufacturer shall state the free air area, type and location of grilles upon which the ratings are based. He shall also state the distance from the bottom of the radiator to the top of the outlet grille which shall be designated as the stack height.

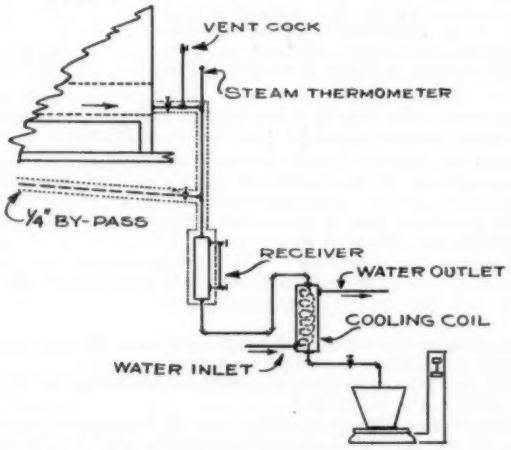

FIG. 2—TEST APPARATUS WITH CONDENSATE COOLER

33. **Sources of Error in Radiator Testing.** The major sources of error in addition to accidental mistakes are as follows:

1 Entrained water brought into the radiator with the steam.
2 Too large a receiver.
3 Loss of condensate during the process of collecting and weighing, either by flashing into steam when pressure is released, or by evaporation from hot water surfaces to the air. This loss may be eliminated by means of a condensate cooler. (See Fig. 2.)
4 Incomplete venting of air from radiator.
5 Excessive air currents within the test booth.
6 Wet and/or insufficient insulation.

No. 900

FLOW OF STEAM THROUGH ORIFICES INTO RADIATORS

By STERLING S. SANFORD[1] *(MEMBER)*, AND CARL B. SPRENGER[2] *(NON-MEMBER)*, DETROIT, MICH.

THE "radiation" installed in buildings must emit sufficient heat to maintain the desired temperature in the coldest weather. For higher outside temperatures, the "radiators" will be of greater capacity than required, and overheating will result unless the heat output is reduced. The amount of heat delivered by a steam "radiator" can be reduced by any one of the following three methods:

1. By lowering the pressure and thus the temperature of the steam.
2. By admitting steam to the "radiator" intermittently and thus reducing the length of time it is filled with steam.
3. By restricting the flow of steam into the "radiator."

In using the last method, the rate of flow of steam into a "radiator" can be controlled by means of (1) a restriction of variable area such as is obtained by throttling the radiator valve, or (2) a fixed restriction, such as a fixed orifice, a variable flow being obtained by varying the pressures on the two sides of the orifice. This paper deals with the last-named method of controlling heat output, that is, by the use of a fixed orifice in the "radiator" inlet.

The use of a fixed orifice makes it possible to reduce the heat output of a "radiator" to any desired extent. The "radiator" may be kept full of steam when the maximum heat output is needed, by adjusting the pressure on the inlet side of the orifice so that it is sufficiently above that in the "radiator." In mild weather the flow of steam into the "radiator" may be reduced to such an extent by reducing the difference in pressure between the two sides of the orifice that only a small part of the "radiator" is heated. Between these extremes, the pressure differential may be adjusted to give the flow of steam required for weather conditions encountered at any time during the heating season.

"Radiator" orifices are installed in various ways. Fig. 1 shows three methods of application. At *A*, a disc containing a straight-sided orifice is inserted in

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Presented at the Semi-annual Meeting of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, Swampscott, Mass., June, 1931, by S. S. Sanford.

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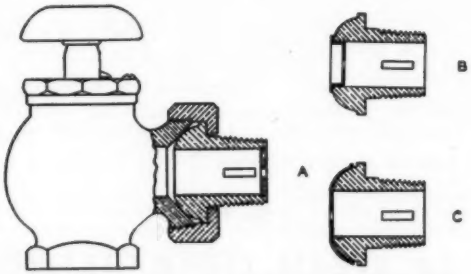

FIG. 1. DIFFERENT TYPES OF ORIFICES SHOWING METHOD OF INSTALLATION IN RADIATOR VALVE

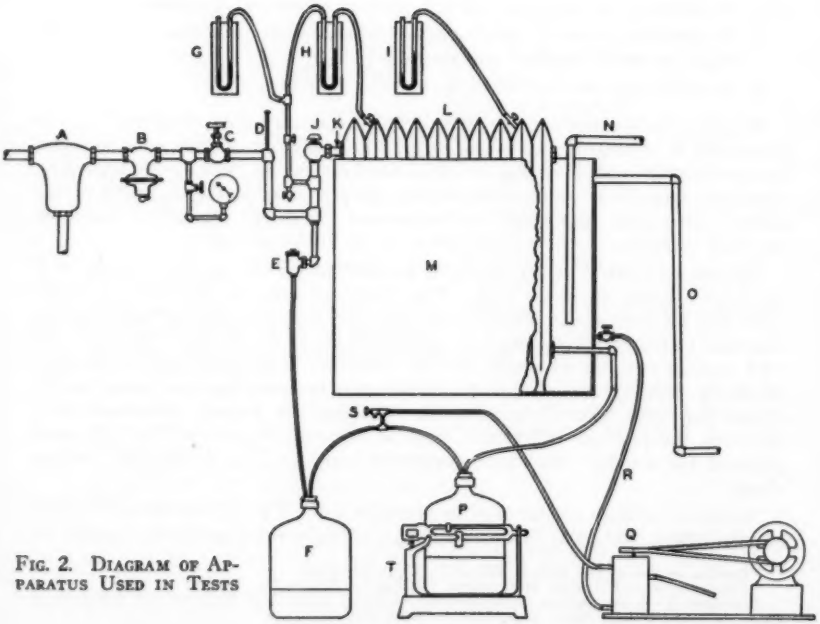

FIG. 2. DIAGRAM OF APPARATUS USED IN TESTS

a recess cut in the "radiator" end of the valve tail-piece. The orifice discs of this type used in the tests described were 3/32 in. thick. The orifice at *B* is in a copper cup-shaped stamping 0.023 in. thick and is inserted in the union end of the tail-piece. The orifice at *C* is in a brass stamping 0.019 in. thick shaped to form a gasket in the union.

"Radiator" orifices have been used in steam heating systems for many years, yet so far as the authors know, no reliable data on the flow of steam through them have been published. Such data as are available are for the flow of steam through orifices installed in straight runs of piping and in general are for higher pressures or for larger orifice sizes than those used for "radiators." The tests described in this paper were made to determine the flow through different sizes of orifices under various pressure conditions for controlling temperature in buildings.

DESCRIPTION OF APPARATUS AND TESTS

The apparatus used in making the tests is shown in Figs. 2 and 3. The apparatus indicated in Fig. 2 is as follows:

A—A steam separator for removing entrained moisture from the steam. The separator was dripped through a special float trap.

B—A ¾-in. pressure reducing valve used for reducing the pressure of the district steam supply from 30 lb per square inch to 3 lb per square inch gage.

C—A ¾-in. globe valve used for throttling the steam to the desired pressure ahead of the orifice.

D—A glass-stem thermometer inserted in a well in the steam pipe.

E—A ½-in. thermostatic trap for dripping the steam pipe ahead of the radiator valve.

F—A bottle with tight connections for collecting the drip from the trap during sub-atmospheric pressure tests.

G—A mercury manometer giving the pressure ahead of the radiator valve.

H—A mercury manometer giving the pressure drop through the radiator valve and orifice.

I—A mercury manometer giving the pressure in the radiator.

J—A ¾-in. standard radiator valve.

K—The valve tail-piece in which the orifice was installed.

L—A 10-section, 2-column radiator, 32 in. high.

M—A galvanized iron tank filled with cooling water in which the radiator was immersed.

N—The cold water supply.

O—The overflow for the tank.

P—The bottle in which the condensate was collected and weighed.

Q—A motor-driven laboratory air pump capable of maintaining a vacuum of 20 in. of mercury in the radiator.

R—The water supply for the air pump.

S—A needle valve for regulating the vacuum in the radiator.

T—Platform scales reading by ounces to 50 lb.

The apparatus shown in Figs. 2 and 3 was used for the sub-atmospheric pressure tests. The same apparatus was also used for the atmospheric pressure tests, except that the vacuum pump was disconnected, the connection between

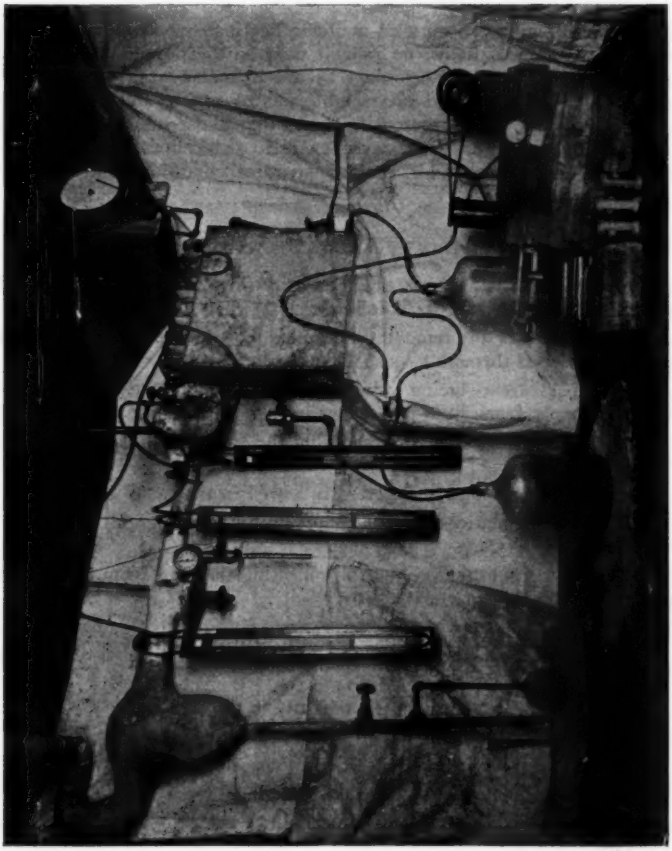

FIG. 3. TEST APPARATUS

bottles *F* and *P* eliminated, and the "radiator" and bottles *F* and *P* vented to the atmosphere.

The data presented in Figs. 4 to 10, inclusive, are from tests with the types of orifices designated as *A* and *B* in Fig. 1. The orifice plate marked *A* was a brass disc 3/32 in. thick and 13/16 in. in diameter. The orifices were drilled 1/64 in. undersize in a drill press and reamed by hand to the desired size. All burrs were removed. The diameters of the orifices tested were 3/32, 7/64, ⅛, 5/32, 11/64, and ¼ in. The plate marked *B* in Fig. 1 was a copper stamping

0.023 in. thick, with the orifice drilled and reamed as for *A*. Three sizes were tested, ⅛, 11/64, and ¼ in. in diameter. To prevent leakage around the edge of the stamping, it was soldered to the tail-piece.

Tests were made with atmospheric pressure in the "radiator" and also with vacuums of 4, 8, 12, 16, and 20 in. of mercury. The atmospheric-pressure tests were made on all of the orifices. The sub-atmospheric pressure tests were made with the ⅛, 11/64, and ¼-in. orifices only. At each pressure, tests were made with pressure drops through the orifice of ½, 1, 2, 4, and 6 in. of mercury by changing the initial pressure ahead of the radiator valve.

The flow of steam through the orifice was determined by weight. This was obtained by condensing the steam in a "radiator" surrounded by cold water and collecting the condensate in a bottle on platform scales. With few exceptions each test ran for one hour. During each test the pressures on the inlet side of the "radiator" valve and in the "radiator" were held as nearly constant as was possible. Any visible deviation from the test pressure was corrected immediately. During the tests at atmospheric pressure, the "radiator" pressure was kept constant by venting the "radiator" to the atmosphere. During the sub-atmospheric pressure tests, the vacuum in the "radiator" was held constant by manipulation of the needle valve *S*, varying the amount of air admitted to the air pump. The pressure ahead of the "radiator" valve was held constant by manipulation of the globe valve *C*.

In starting a test, the pressures were held constant for about 15 minutes before taking the initial scale reading. This allowed the temperature and other conditions to reach equilibrium. Pressure and temperature readings were recorded every 10 minutes. The time required to accumulate equal weights of condensate in the bottle *P* was recorded as the test progressed. If the time varied appreciably indicating non-uniformity of flow, the test was continued until an hour of uniform flow was obtained. In checking the time, the weight was moved along the scale beam in increments corresponding to the condensation taking place in 10 min. The electric bell attached to the scales (Fig. 3) assisted in making this check.

No trap was used on the "radiator" discharge because the condensation rate of the water-cooled "radiator" was sufficient to prevent steam from reaching the outlet. During the sub-atmospheric pressure tests, bottles *F* and *P* were connected as shown so that the same vacuum existed in them as in the "radiator." The discharge of trap *E* was thus always at a lower pressure than its inlet.

Moisture in the steam was supposedly removed by the separator *A*, and any which remained was evaporated by the reduction in pressure from 30 to 3 lb per square inch in passing through the reducing valve *B*. The piping was well covered, the distance from the reducing valve to the orifice was small, and the piping was dripped through a trap ahead of the orifice. Nevertheless, it was feared that there might be moisture in the steam, and since the flow was being determined by weight, it was very important not to have any moisture in the steam passing through the orifice. In order to determine if slightly superheating the steam by adding heat to it would change the results, a coil of ¼ in. copper tubing about 15 in. long was inserted in the steam supply between valves *B* and *C*. The coil was heated with a blow torch sufficiently to raise the tem-

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perature of the steam one or two degrees. After a series of tests which gave the same result as without the heating operation, the tubing was removed and the tests continued. Thermometer *D* indicated approximately the saturation temperature in all tests. When a high vacuum was being carried at this point, a superheat of one to two degrees was observed.

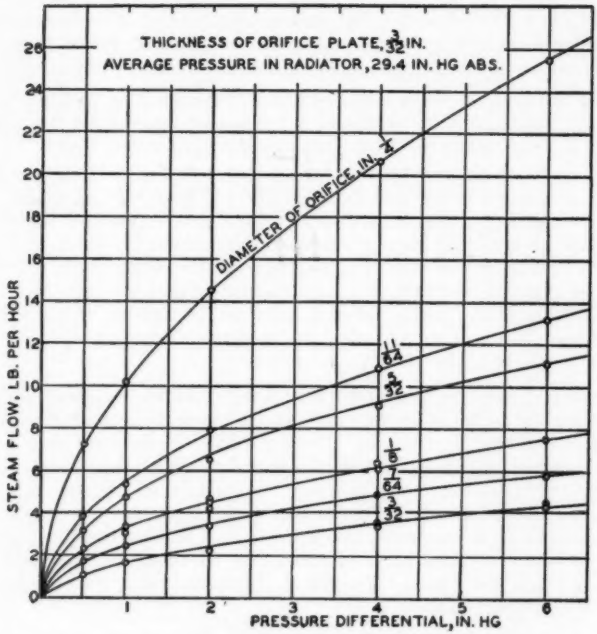

FIG. 4. FLOW OF STEAM VS. PRESSURE DROP THROUGH ORIFICE FOR ORIFICES IN A PLATE 3/32 IN. THICK

Two makes of "radiator" valves were used during the tests but no difference in the results could be detected.

RESULTS

The results of the tests on the orifices in the 3/32-in. thick plates with atmospheric pressure in the "radiator" are indicated by the curves in Figs. 4 and 5. The orifice plates were installed in the "radiator" end of the tail-piece. Fig. 4 shows the relation between flow and pressure drop through the orifice. It is well known that flow is not directly proportional to pressure drop. As shown by this figure, when the pressure drop through the orifice is low, small changes in pressure drop affect the flow to a greater extent than do the same changes in the region of high pressure drop.

Fig. 5 shows the relation between flow and cross-sectional area of orifice. It should be noted that the flow is not exactly proportional to the area. As the area of the orifice increases, the rate of increase of flow decreases.

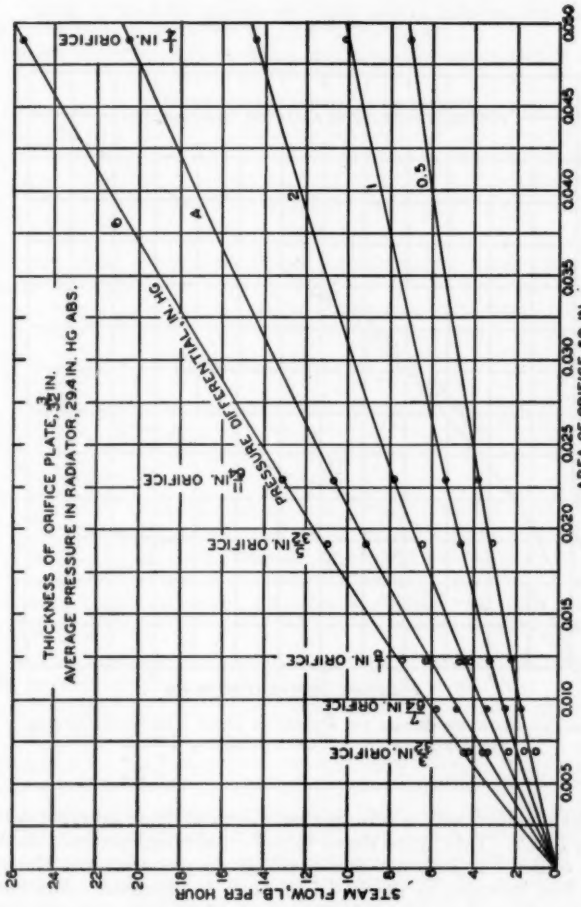

FIG. 5. FLOW OF STEAM VS. AREA OF ORIFICE FOR ORIFICES IN A PLATE 3/32 IN. THICK

The results of the tests on the orifices in the copper stampings 0.023-in. thick installed in the union end of the tail-piece with atmospheric pressure in the "radiator" are given by the curves in Figs. 6 and 7. The flow characteristics are similar to those for the thicker orifices except that the flow is slightly reduced.

Fig. 8 shows the relation between flow and orifice area for the orifices in the 3/32-in. thick plates when a vacuum of 20 in. of mercury existed in the "radiator." It should be noted that the flow was considerably reduced as compared with that at atmospheric pressure. The complete results for the two types of orifices with vacuums of 4, 8, 12 and 16 in. of mercury in the "radiator" are not included in this paper on account of lack of space.

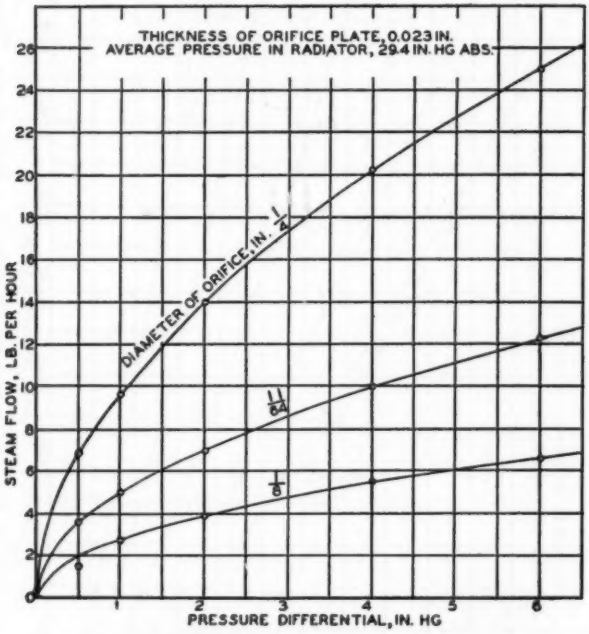

FIG. 6. FLOW OF STEAM VS. PRESSURE DROP THROUGH ORIFICE FOR ORIFICES IN A PLATE 0.023 IN. THICK

The results obtained with a 6-in. pressure drop at these different conditions are given in Fig. 9. This figure shows the effect of initial pressure on the flow through an orifice. It indicates that for a given pressure drop through the orifice, the flow increases as the initial pressure is increased. The curves in Fig. 9 are for the orifices in the 3/32 in. thick plate with a pressure drop of 6 in. of mercury through them and are taken from the tests with pressures in the "radiator" varying from atmospheric to a vacuum of 20 in. of mercury.

The flow through a nozzle is given by the Moyer formula,

$$W = \frac{AP^{0.97}}{60} \qquad (1)$$

which is approximately the same as Grashof's formula,

$$W = 0.01654AP^{0.9696} \quad (2)$$

where—

W is the weight of dry steam discharged in pounds per second,

A is the area of the orifice in square inches, and

P is the initial absolute pressure in pounds per square inch.

This formula holds only when the ratio of the final pressure to the initial pressure is less than 0.58. When the ratio of the final pressure to the initial pressure is greater than 0.58 the result given by the formula must be multiplied by a factor to obtain the actual flow. This factor is a function of the pressure ratio. The values for it as determined by Rateau [3] are given by the upper curve in Fig. 10.

The flow through an orifice is less than through a nozzle and does not reach a maximum when the ratio of the final pressure to the initial pressure is 0.58. The flow continues to increase as the final pressure is reduced below 58 per cent of the initial pressure. The formula just discussed can also be used for calculating the flow through an orifice if the proper coefficients are used with it. The lower curve in Fig. 10 gives the coefficients for an orifice in a thin plate as determined by Rateau. The orifice which he tested had a diameter of 20.12 mm (0.792 in.) and was installed in a pipe having a diameter of 60 mm (2.4 in.). The discharge side of his orifice was cut away, making it a true thin-plate orifice.

The individual points on the curve sheet are from the tests reported in this paper. It will be noted that the points obtained from the tests on the thin copper orifices agree closely with the curve given by Rateau, whereas the flow through the thicker orifice is somewhat greater, approaching that through a nozzle as the ratio of orifice diameter to thickness of plate decreases. The point for the pressure ratio of 0.405 was determined from a special test.

During the various tests reported in Figs. 4 to 8, the barometric pressure varied between the limits of 29.15 and 29.54 in. of mercury. For most of the tests it averaged 29.4 in. The average during the series of tests shown by Fig. 8 was 29.3 in.

Table 1 shows the results of a series of tests made to determine the effect of moving the orifice from one end of the valve tail-piece to the other. In every case the flow of steam was greater with the orifice installed in the union end of the tail-piece. The data in columns 2 and 3 are from Figs. 4 to 7. All of the tests were run with atmospheric pressure in the "radiator." The average barometric pressure was 29.4 in. of mercury and the individual results were corrected to this pressure by using the curves in Fig. 9. During these tests the thin orifice plate was held in place by soldering it to the tail-piece.

The increased flow obtained with the orifice installed in the union end of the valve tail-piece can be explained by the probable formation of a region of reduced pressure in the tail-piece surrounding the vena contracta. This would

[3] Experimental Researches on the Flow of Steam Through Nozzles and Orifices, A. Rateau, D. Van Nostrand Co., 1905.

occur if the flow filled the tail-piece at its discharge end. The action of the fluid steam would then partly evacuate the space formed. The pressure surrounding the vena contracta being less than the pressure in the radiator, the

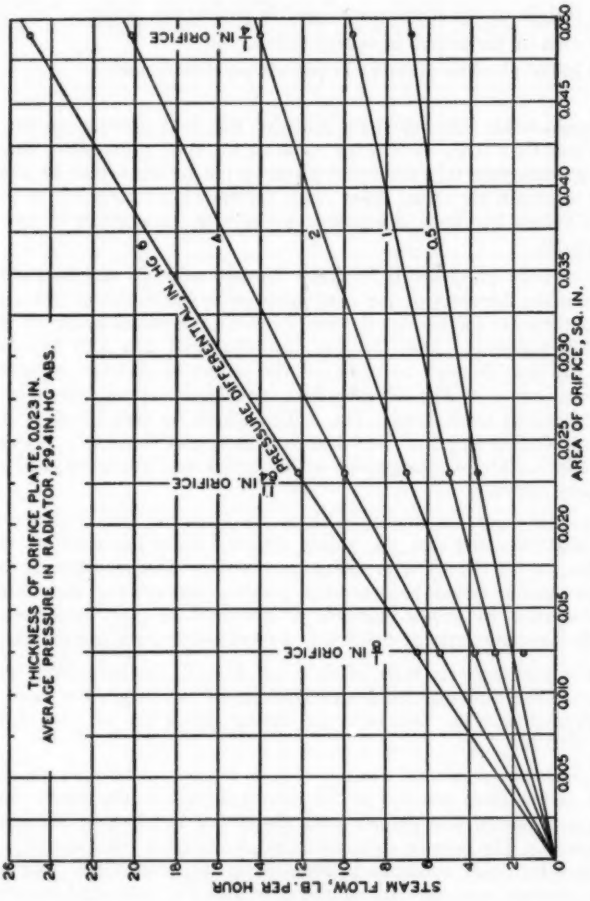

FIG. 7. FLOW OF STEAM VS. AREA OF ORIFICE FOR ORIFICES IN A PLATE 0.023 IN. THICK

flow would correspond to this reduced discharge pressure rather than to the pressure in the "radiator."

Table 2 shows the results of a series of tests made to determine the effect of the thickness of the orifice on the flow of steam. All of these tests were run with the orifices in the "radiator" end of the tail-piece. The data in the first column are for the copper stamping soldered in the end of the tail-piece.

TABLE 1. EFFECT OF POSITION OF ORIFICE ON STEAM FLOW [a]

Pounds per Hour Through ¼-in. Diameter Orifice

Pressure Drop Through Orifice (In. Hg)	Orifice in a Plate 3/32 In. Thick		Orifice in a Plate 0.023 In. Thick	
	Orifice at Union	Orifice at "Radiator" End	Orifice at Union	Orifice at "Radiator" End
1	10.8	10.2	9.7	9.2
2	15.4	14.4	14.1	13.7
4	22.6	20.6	20.2	19.0
6	27.1	25.6	25.0	24.1

[a] "Radiator" pressure, 29.4 in. of mercury, absolute.

The data in the second column are from the results given in Figs. 4 and 5. The data in the third column are from an orifice plate ¼ in. thick made for this test. These tests, which were run with atmospheric pressure in the "radiator" and corrected to 29.4 in. of mercury, absolute, show clearly that the flow increases with the thickness of the plate, indicating that the flow through a thick plate approaches that through a nozzle. Other investigators have found that for air, the maximum flow is obtained when the thickness of the plate equals the diameter of the orifice.[4]

APPLICATION

The tests demonstrate that it is possible accurately to control the flow of steam into a "radiator" by controlling the pressures on the two sides of an orifice located in the "radiator" valve. In a heating system, however, the pressure ahead of every "radiator" valve is not the same because of the pressure drop in the piping, and therefore the flow of steam through the orifices into the "radiators" nearest the source of supply will be greater than into the more distant "radiators." This inequality of flow can be minimized by using small orifices so that the pressure differential required across the orifice to produce a given flow will be large compared with the pressure drop through the piping.

It is not considered good practice to design a heating system for a total pressure drop through the piping of more than 8 oz. In a 40-story building in which the piping is designed for a pressure drop of 1 oz per 100 ft, the maximum difference in pressure between the nearest and farthest "radiators" might be of the order of 1 in. of mercury. This difference would occur only in extreme weather when the heating system is operating at maximum capacity. At other times, the pressure drop would of course be reduced because of the reduced flow of steam.

TABLE 2. EFFECT OF THICKNESS OF ORIFICE PLATE ON STEAM FLOW [a]

Pounds per Hour Through 11/64-in. Diameter Orifice

Pressure Drop Through Orifice (In. Hg)	Thickness of Plate, 0.023 In.	Thickness of Plate, 3/32 In.	Thickness of Plate, ¼ In.
1	4.8	5.4	6.4
2	7.0	7.8	8.6
4	9.7	10.7	11.9
6	12.0	13.2	14.1

[a] "Radiator" pressure, 29.4 in. of mercury, absolute.

[4] Steam and Gas Turbines, A. Stodola, Vol. I, pages 113-114. McGraw-Hill Book Co., Inc., 1927.

The allowable pressure drop through the orifice is limited by the noise, which must not be loud enough to annoy room occupants. It has been found by experience that a pressure drop through the orifice of 6 in. of mercury can be used without causing annoyance. If the orifices are made of such a

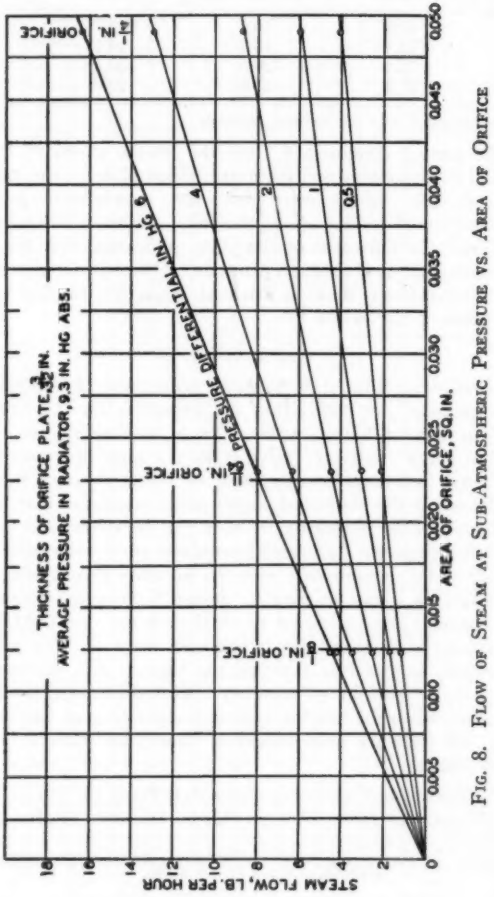

FIG. 8. FLOW OF STEAM AT SUB-ATMOSPHERIC PRESSURE VS. AREA OF ORIFICE

size that a pressure differential of 6 in. of mercury is required to keep the "radiators" full of steam under normal conditions, then the sizes should be as given in Table 3. These sizes are based on atmospheric pressure in the "radiator" and orifice plates 3/32 in. thick located in the "radiator" end of the valve tail-pieces. The condensation rate of the "radiator" is assumed to be ¼ lb of steam per hour per square foot of equivalent "radiation." If one of

the other types of orifices is used instead, this table can be revised from the data given in this paper.

If the total pressure drop in the piping between the first and last "radiators" of a heating system is high, the required flow of steam into the more distant "radiators" can be obtained by using larger orifices in these "radiators." Another means of obtaining the desired steam flow into all of the "radiators"

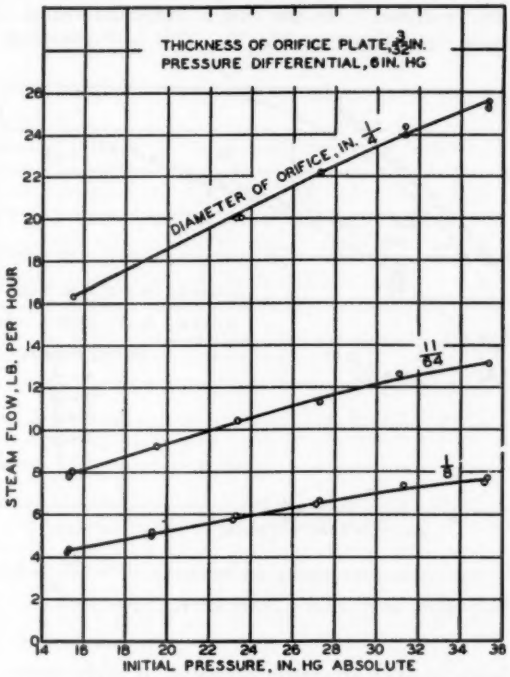

FIG. 9. EFFECT OF VARYING INITIAL PRESSURE WHILE KEEPING PRESSURE DROP THROUGH ORIFICE CONSTANT

is to divide the heating system into zones and to control the pressure in each zone independently.

Table 4 gives the pressure differential required at the orifices to deliver the proper amount of steam to the "radiators" at different outdoor temperatures assuming that a "radiator" full of steam at atmospheric pressure will supply the required amount of heat when the outside temperature is zero and that the orifice sizes are as given in Table 3 and atmospheric pressure is maintained in the "radiators." This table was taken from Fig. 4 and is based on the assumption that if a 6 in. differential pressure gives the proper amount of steam at zero degrees, then the amount of steam required at any other temperature is proportional to the difference between 70 F and the outdoor temperature.

The pressure in the heating system is usually controlled by adjusting a pressure reducing valve or a throttle valve in the steam main. This control valve must be set for a pressure sufficient to take care of the pressure drop in the piping between the control valve and the "radiators" in addition to the required pressure drop through the orifices. Table 4 shows that a very small differential pressure across the orifices must be used in mild weather. If the heating system is equipped with a vacuum pump, sub-atmospheric pressure can be carried in the "radiators." In that case a larger differential must be used to obtain the same steam flow (see Fig. 9). This is of advantage in that the

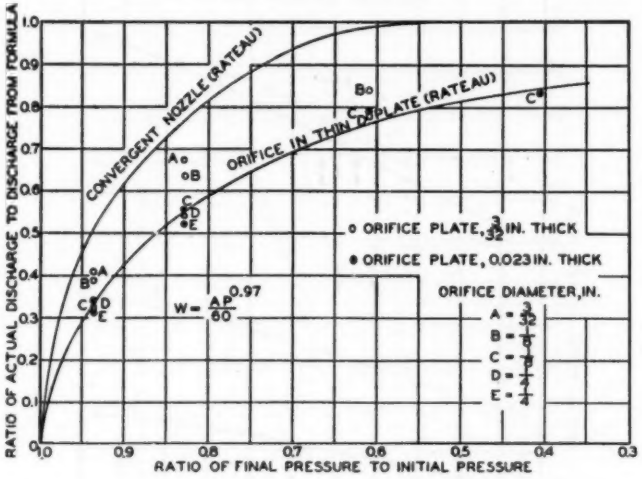

FIG. 10. VALUES OF COEFFICIENT FOR FORMULA, $W = \frac{AP^{0.97}}{60}$ SHOWING COMPARISON OF AUTHOR'S TESTS WITH CURVES BY RATEAU

flow can probably be more accurately regulated, as slight errors in adjusting the pressure will not affect the flow through the orifices as much as when smaller differentials are used.

The data presented in this paper have been based on the assumption that the system would be controlled by maintaining a constant pressure in the "radiators" and by varying the pressure in the steam mains. In vacuum systems, if desired, the steam pressure can be kept constant and the vacuum in the "radiators" varied to provide the required differential.

While the practice of dividing a building heating system into zones is desirable from the standpoint of control, it is not absolutely essential to the successful operation of an orifice system. One building having a gross volume of over 20,000,000 cu ft and containing about 270,000 sq ft of heating surface with orifices in the "radiators" is operating successfully with the heating system divided into three zones. In another case, a 46-story building with

TABLE 3. SIZES OF "RADIATOR" ORIFICES

Orifice Diameter (In.)	Size of "Radiator" (Sq Ft)	Orifice Diameter (In.)	Size of "Radiator" (Sq Ft)
3/32	18	3/16	61
7/64	24	13/64	70
1/8	30	7/32	80
9/64	37	15/64	91
5/32	44	1/4	102
11/64	52		

orifices in the "radiators" is operating successfully with one control valve for the entire building.

The "radiators" in an orifice heating system can be quickly filled with steam in the morning at the beginning of the warming-up period by using a sufficiently high differential pressure, so that the length of the warming-up period need not be appreciably longer than with an ordinary heating system.

A heating system equipped with "radiator" orifices has several advantages:

1. The building temperature can be controlled.
2. The amount of steam used for heating is reduced.
3. The building temperature control is in the hands of the building management, where it can be properly supervised.
4. Excessive opening of windows and the resulting loss of heat is avoided.
5. The cost of making and installing orifices is very low. (Those shown in Fig. 1 at *B* and *C* are easy to install and the stampings are inexpensive to make in quantities.)

It is desirable to use manometers instead of pressure gages in adjusting the pressure at the control valve as accurate adjustment of pressures is essential to the successful operation of an orifice system. Orifice systems should be installed only in those buildings where there is a member of the operating staff available who will give close attention to adjustment of pressures. An electrical thermometer system giving temperatures throughout the building is of assistance in operation.

CONCLUSIONS

The flow of steam through thin-plate orifices installed in the unions of "radiator" valves is substantially the same as reported by Rateau for a thin-

TABLE 4. REQUIRED PRESSURE DIFFERENTIAL ACROSS ORIFICES FOR DIFFERENT OUTDOOR TEMPERATURES [a]

Outdoor Temperature (Degrees Fahrenheit)	Pressure Differential (In. Hg)	Outdoor Temperature (Degrees Fahrenheit)	Pressure Differential (In. Hg)
0	6	40	1.2
10	4.5	50	0.5
20	3.2	60	0.1
30	2.1		

[a] Atmospheric pressure in "radiator"; room temperature, 70 F.

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plate orifice in a straight run of pipe. The flow can be calculated from the formula $W = \frac{AP^{0.97}}{60}$ by multiplying the result by a factor obtained from Rateau's curve for thin-plate orifices. For thicker orifices the results indicate a higher rate of flow than given by this curve.

"Radiator" orifices can be inexpensively made and installed. When an orifice system is properly operated, it is possible to obtain highly satisfactory control of building temperature, resulting in economy in the use of steam for heating. Table 5 shows the savings made in two large Detroit office buildings by installing orifices in the "radiator" valves. Both of these buildings were very carefully operated before the installation of orifices and the steam consumption at that time was about the minimum that could be obtained without using some type of control. The savings reported, therefore, are savings due alone to the installation of orifices. The steam consumption reported is for heating only and does not include steam for water heating and other purposes. About 11 per cent of the space in building *B* is occupied by a bank which is air-conditioned, so that the steam consumption of that part of the building was not affected by the installation of orifices.

TABLE 5. RESULTS OBTAINED BY INSTALLING ORIFICES IN "RADIATORS"

	Building *A*		Building *B*	
	Before Installing Orifices	After Installing Orifices	Before Installing Orifices	After Installing Orifices
Period covered	1-31-29 to 5-31-29	1-31-30 to 5-31-30	11-30-29 to 3-31-30	11-29-30 to 3-31-31
Average temperature, degrees Fahrenheit	42.7	44.1	29.4	31.1
Degree-days (65 degree Fahrenheit base)	2736	2572	4308	4136
Steam for heating, pounds	39,040,400	32,958,860	21,818,300	17,558,800
Steam for heating, pounds per degree-day	14,300	12,810	5,065	4,240
Steam saved by installing orifices, per cent	.....	10.4	.....	16.3

	Type	"Radiation" (Sq Ft)	Net Heated Space Inside Building (Cu Ft)
Building *A*	Office Building	267,560	16,437,510
Building *B*	Office Building	93,190	6,858,360

ACKNOWLEDGMENT

The tests reported in this paper were made in connection with the work of The Detroit Edison Co. in instructing its steam heating customers in the economical utilization of steam for heating. The authors are indebted to C. W. Signor and E. T. Smith of the Central Heating Department, who conducted the tests and who contributed many helpful suggestions.

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DISCUSSION

J. A. DONNELLY (WRITTEN): Much reliable data on this subject are available, but care should be taken to distinguish between the flow through thin-plate orifices and that through thicker plates in which the orifices approach or reach the construction of short tubes, where flow is always of greater quantity.

In removing the burrs it is also difficult, unless great care is used, to leave the edges of the orifices sharp and thus avoid the conditions of rounded entrances and exits, which will materially increase the flow.

The authors make some general and sweeping statements as to what has been, or may be considered *good design* in relation to maximum allowable pressure drop in piping systems. It should be borne in mind that the proper application of orifices may materially modify our former pipe size practice. For instance, the use of compound orifice systems, where an additional set of plates are used in the riser connections to compensate for steam main friction, as reported by Seiter, Calvert and Hunter in the *Proceedings of the National District Heating Association,* will give the stamp of *good design* to steam main sizes of comparatively high drop in pressure.

If a simple orifice system is to be used, it is, as the authors state, essential that the steam mains be larger and a higher drop be used through the radiator orifices. One of the limitations of the latter is the noise produced.

It is further stated that a drop corresponding to 6 in. of mercury pressure is allowable. It is not clear, however, whether this experience was obtained solely

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with a cast-iron radiator which would muffle the noise, or whether some experience was had with sheet copper, brass and other light metal constructions which would broadcast the noise. Further investigation may also show that some of the small orifices used with high drop on small radiators may give trouble with dirt accumulation or stoppage.

With reference to the desirability of *zoning* and to the economies produced, I am of the opinion that we need most of all a uniform and an accurate standard of test comparison. This I would suggest as *pounds of steam per hour, per heating degree, per million Btu ascertained heat loss.* Calculated heat losses are rather a poor substitute, and I would suggest the following method of test or research for ascertaining actual heat losses.

Select typical rooms on the several sides of a building, disconnect the radiators and elevate them slightly so that the condensation may be collected. Feed the necessary steam into them either by hand or automatically so that the room temperature is properly maintained, and from the data collected the actual heat losses eventually may be accurately ascertained.

The most difficult phase of the problem is probably the varying effect of wind velocities and sunshine. The ultimate solution of these variables will perchance point to the desirable aspects of *zoning.*

As a side line to this research, I would also suggest variable window construction, outside wall insulation, weatherstripping and calking of window frames.

A. A. ADLER (WRITTEN): In the design of all systems which involve fluid flow the necessary and sufficient condition is that the circulating head must equal the sum of all resistance heads. In the case of hot water heating systems it is a simple matter to adjust the fluid resistance by suitable choice of pipe sizes and number of fittings to insure balance among the various circuits. In addition, hot water heating systems have an inherent tendency to balance themselves even though the resistance has not been accurately predetermined since a slowing up of circulation causes greater cooling and so increases the temperature difference between flow and return which, accordingly, tends to increase the circulating head to the end of augmenting the flow. Should the opposite condition prevail, the temperature difference between flow and return diminishes and so slows up the flow. Moreover, the fluid is homogeneous and flows in a constant direction. Accordingly hot water heating systems when once properly installed remain so for all time and at all loads unless obvious elements enter to disturb the original relation between circulating and resisting heads.

This splendid fundamental relationship is not strictly true for steam heating systems. They carry water, steam and air in varying proportions in their normal and abnormal operating conditions. To add to the disturbance of flow, the water, steam and air do not always flow in the same direction. Accordingly, the estimated circulating and resisting heads do not always remain in balance even though they be correct at certain loads. This effect is most noticeable at light loads.

As is generally true in all process work that control of the final effect requires satisfactory control of the elements which enter into the process. In the design of steam systems, the designer is not in complete control of all elements that affect subsequent flow of steam, water and air. The customary practice in design of heating systems is to furnish the fitter with a diagramatic layout which not infrequently is subject to change in the actual installation.

Accordingly, fixed orifices based on original assumptions may not be entirely satisfactory in making the needed correction.

In some forms of orifice distribution systems the pressure in the distribution is relatively high so that the pressure drop is negligable and therefore the area of the orifice is directly proportional to the heat emission of the radiator. This calls for a small orifice, usually round, which is likely to hold up water and dirt and because of the

high pressure drop across the orifice, is subject to possible erosion due to the high velocity through the orifice. A change in one or more of these elements alters the quantity of steam passed to the radiator.

The better plan is to have distribution at low pressure with low pressure drop across the orifice, thereby increasing the area of orifice and decreasing noise and the probable erosion in operation.

Another element which affects the computation of orifices is the steam passed in horizontal mains and vertical risers. In the horizontal distribution only frictional resistance need be reckoned with whereas in vertical distribution the work of elevating the steam must be considered in balancing the distribution system. While these factors may be computed beforehand, some inherent adjustments in a practical system are eminently desirable.

With a view of exercising control of the variables and to give flexibility to unforeseen changes in installation, operating conditions, extensions to the system, etc., a system which attempts this might be of general interest.

The system consists of an adjustable orifice at the base of each riser, an adjustable orifice in the inlet valve, and an automatic control varying the quantity admitted to the zone or system.

The valve at the base of the riser consists of a drum rotatable within the valve body. It has a rectangular opening, the vertical dimension being the nominal diameter of the pipe and the other dimension two-thirds of this. The drum diameter is twice the width of the opening so that in 60 deg the valve is capable of giving any opening between wide open and shut. Graduations on the valve body show the amount of the opening. The steam, water and air in passing through the valve are disturbed as little as possible, and the actual flow can be predicted from test data or computation. In addition, there are two openings on opposite sides of the drum so that the pressure drop is made in two stages thus dividing the velocity acquired between them. Because of the symmetry the flow to and from the valve is believed to be more uniform than it would be for non-symmetrical forms of orifices.

On the radiator inlet valve the orifice is placed under the seat of the valve and is of a quadrant form symmetrically placed. The area is adjustable through the hollow valve stem.

When first installed all orifices are wide open and the system is allowed to operate as an ordinary gravity or vacuum system. After all dirt is washed to its final lodgment the system is balanced by appropriate adjustment. This is made by direct computation of the resistance.

Should any one choose to over- or underheat, any radiator adjustment is possible while the system is in operation.

The valve adjustments are made with special tools so that they cannot be tampered with. Such adjustments require more experience than the usual fitter possesses and accordingly it is deemed best to leave them to those trained in this particular work.

WEBSTER TALLMADGE (WRITTEN): This paper is one that has been much needed for a better understanding of the important and somewhat overlooked problem of economy in building heating. The paper is along basic lines and deserves favorable discussion and comment. The following discussion is intended to augment and amplify the information rather than contradict the items presented.

There are many ways of installing radiator orifices not shown in the paper. For example:

(1) They may be put ahead of or in the valve seat. When so placed they are not always stable in performance so that the accuracy is not dependable.

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(2) In many cases in actual commercial buildings, orifices must be placed in the end of right and left, or standard right hand, threaded pipe nipples.

(3) For cast-iron radiation the orifices are placed preferably in a wall in the outlet of a special bronze radiator bushing.

(4) Union type orifices fit only the particular shape of union face for which they are made. This may vary not only for one maker's valve but also for valves from different years from the same maker. The shape used by each maker is different from the others. Encountering this in many tall buildings, studies were made to accumulate data on the possibility of obtaining a shape that would be universal to all unions. Orifices of this type are not only universal but have the stability and accuracy of a nozzle as compared with the variable performance with thin-plate orifices described in the following.

Several other methods of installing orifices have been used, but because they either leak steam around the orifices causing inaccuracy, are used with low differential, or are inaccurate in some other way, they should not be described, for this paper has the spirit of accomplishing high grade accuracy throughout which is the essential thing.

In manufacturing orifices for the wide range of conditions encountered in actual commercial application thin-plate orifices were found to be not as reliable as those in which the orifice is made in a wall, the metal of which is about $\frac{3}{16}$ in. thick. The thick-wall or tube type is easier to make accurately. It is easier to keep the metal true and therefore it is more reliable in its actual performance in the building heating systems. Frequently the question is asked why not make the orifice with rounded inlet to ease the flow. This would give the reverse of what is wanted, namely, accuracy with high resistance to obtain good distribution.

The foregoing does not mean that the use of fractionally sized holes is all that is necessary. It is not only necessary to make an accurate hole but to choose accurate sizes for each size of radiator. For illustration, an orifice that is accurate for a 45-sq ft radiator is not sufficiently accurate for a 46-sq ft radiator.

In 1919-1920 a group of buildings was operated by varying the steam consumption according to the weather by varying the vacuum. Again in 1923 another group was found to be operating at as high as 20 in. vacuum on mild days and at positive pressure on cold days. Orifices were installed and then it was discovered that one building was performing poorly (it always had performed poorly although it was the newest one in the group). The trouble was caused by a multiplicity of small leaks almost impossible to find. Air leaked into the space in the pipes where the steam should have been so that the quantity of steam admitted to the system had to be greatly increased. This resulted in over-supplying the easily reached parts to get heat to those parts suffering from poor distribution.

It is therefore of little value to consider accurate orifices on systems that may be erratic due to air in the steam pipes. If the air leak is in a return pipe instead of a steam pipe, the pressure differential will be reduced across all of the orifices in the radiators being served by this particular return pipe and less steam will flow into the radiators through the orifices. Expansion joints in risers in tall buildings are a particularly annoying source of such trouble. It is therefore a less complicated problem to control the steam supply only, to a building of orifice distribution, particularly if it is zoned. The use of atmospheric pressure in the radiators and return piping permits this and eliminates troubles from leaks. All of the zones can then be on a common return piping system with the returns from zones at the top and bottom northeast and southwest, etc., all on a common gravity atmospheric return with no air valves. The air and condensate are both heavier than steam and therefore flow to the bottom of the system and the air flows naturally out of a plain open vent. In any and all high grade systems either sub-atmospheric or having atmospheric gravity return the pipes must pitch or there will be poor distribution of the

steam, noise and in some cases serious damage to such devices as trap elements from water hammer causing continual wastage of steam thereafter.

By referring to Figs. 4 and 6 in the paper, the curves of flow through the orifices, it will be noticed that the pressures at low rates of flow are small. It is therefore not possible to control them, particularly in large buildings having many heating zones with the usual non-technical attendants. For proper control it is necessary to have apparatus that will vary the steam supply according to the curve of flow. These small pressures are, however, surprisingly stable and consistent and for this reason are controlled easily by an electrically-driven, electrically-remotely-controlled steam admission valve operating mechanism. This device compensates for the characteristics of the flow curve and it is extremely sensitive at low pressures. It has a flat regulating feature. If set for 25 per cent flow, the radiators, regardless of size, kind or location, will become 25 per cent hot and continue regardless of turning radiators on or off and regardless of adding or removing radiators. This eliminates what has been termed balancing orifice systems. The valve can be remotely operated from a central control station or it can be automatically operated by a clock, wind, or sun temperature. It will automatically reduce the steam proportionately when warm air leaves the building from the effects of wind on the leeward side or from buoyancy effect of hot air rising to the upper floors and out of the building on a cold day.

We now come to a very interesting point in the discussion of the use of accurate orifices. When these orifices are properly made and the steam supply to them is properly controlled according to the characteristics of the flow through them, it is an extremely simple problem to prevent steam from getting into the return piping. By studying the curves in Fig. 4 it will be noted that at the high rates of flow the upper right end of the curve appreciably flattens. This shows that the pressure changes considerably without greatly changing the flow. It is by the practical application of this fact that permits large (or small) buildings equipped with accurate orifices and the compensated mechanism described to so control the steam supply that radiator return traps are not used. Many buildings are now in regular operation without radiator return traps and with no steam in the returns. The mechanisms are adjusted to keep the maximum return temperature at the bottom of the system below 140 F.

In actual practice it is essential in all building heating systems that the surplus moisture be drained out of the steam either through a separate drip main to one master trap for each zone or a drip trap at the foot of each steam riser. The steam pipes themselves must be pitched and drained to drips because small accumulations of water are swept into elbows or fittings and the water piles up in waves causing noise and high resistance to the steam flow. This trouble is frequently overlooked in the horizontal runs in steam risers at set backs in tall buildings where the condensate falls to the bottom of the horizontal pipe and the current of steam sweeps it into a pile of water in the elbow at the corner where the pipe turns upward again. Points such as this should be dripped. Using a vacuum to clear the lines, so to speak, only increases the velocity and makes the condition worse.

From experiences with thin-plate and thick-plate or nozzle type orifices the indications are that the steam eddies from the valve passages affect the flow through the thin plate. The eddies evidently are elongated at high flow and more nearly circular at low flow. When the outer edge of a circular eddy cuts the edge of the thin orifice aperture there is one rate of flow, and when the center of the eddy reaches the aperture there is another rate of flow. The disturbance in the steam flow entering an orifice may sometimes be oscillating rather than whirling. If the oscillating waves are of short length then the steam flow through the thin-plate orifices is not in proportion to what it would be with longer wave length. This may account for more confidence being placed in the stability of the thick orifices, and also for the

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erratic performance of some jobs with thin-plate orifices which may possibly be caused by eddies or disturbances in the valve passages ahead of the orifice.

In general it seems that the closer the orifice is to the radiator condensing surface, as in the case of the orifice in the wall of the radiator bushing, the more stable the flow.

Moyer's formula is empirical and therefore approximate. It is doubtful if there ever will be a satisfactory formula because the coefficient is evidently a variable. The practical test of orifices and their performance on a number of installations give the only reliable data.

It is interesting to note that there is slightly less pressure inside of the radiator adjacent to the stream of steam leaving the orifice than in other parts of the radiator. At times of high flow there will be a slight suction caused by the friction of the steam against the surrounding air. This suction draws the air into the steam causing modified temperature of the radiator when only partially heated.

The slight noise sometimes noticed in radiators is probably more a function of the organ pipe principle in the radiator itself than of the orifice. This noise has been noticeable on jobs without orifices.

With a zoned system the pipe runs are shorter than in ordinary systems. By feeding the steam mains at the center of gravity of the load, the pressure drop is so reduced that the pipes can be made smaller and be well within good practice for practical distribution. We allow 4 per cent maximum difference in percentage of steam flow to the first and last (or most remote) radiators. It is easily seen that due to the flatness of the flow curve at full-rate flow, the piping pressure drop can be considerable without appreciably affecting the flow. The pipe friction is less under gage pressure than it is under vacuum. Distribution based on this principle is a simplified problem, for the size of the orifice need not be varied for the distance from the supply and for this reason unbalancing is an impossibility. The practice of sizing pipes based on the flow to the last radiator, rather than basing the sizes on pressure drop per 100 ft of run, will be a step forward.

Reducing valves are not sensitive enough for economical control without accurate orifice. Some small hand-controlled jobs show good economy even without orifices, but such jobs are far from practical in large buildings.

Shortening the warming-up period may result in poor distribution, water hammer, with damage to traps, and noise. The ordinary operating moisture is usually so finely divided in mist and so evenly distributed that it goes through the orifice without appreciably affecting the distribution.

It is true that buildings zoned with high-resistance, accurate orifice distribution have lower starting load and heat up quicker due to even distribution of the heat where needed. Low starting load means smaller capitalization for steam generating equipment and a tendency toward better rates from district heating companies.

Individual radiator thermostats may, in some cases, cut down the steam consumption of some rooms as, for example, those overheated by illumination. The additional use of orifices help the economy of individual thermostatically-controlled radiators by preventing an oversupply of steam which would result when the steam valve is opened by the thermostat in response to the chilling effect caused by opening the windows. The universal orifice previously mentioned slips in easier and is self-centering to a greater extent than those that fit the rounded surface of the union and require less time to install.

Orifices should never be put in a down-flow pipe without a strainer ahead of them. It is good practice to put a strainer of not larger than $\frac{1}{16}$ in. mesh in each steam main, particularly on long new lines for pebbles, sand, etc., do get into new underground lines when they are laid. There are few if any cases of a dirt-plugged orifice in an isolated building even on new jobs, but there are cases where the dirt from

pipes caused annoyance. The dirt problem is experienced with all heating devices regardless of whether or not damage is done to orifices, thermostatic radiator valves, radiator return traps, or even reducing valves.

A great deal of credit should be given these writers for they have opened to the heating profession something that should be better understood by everyone.

PERRY WEST: Can Mr. Sanford give us some idea of the comparative saving and the comparative cost as between these zone control systems and ordinary temperature control, with individual controls on the radiators?

MR. SANFORD: Mr. Tallmadge in his discussion suggested that the flow through a thick-plate orifice is likely to be more consistent than through a thin-plate orifice. Our tests on thin-plate orifices gave consistent results.

He also suggested the difficulties encountered in using sub-atmospheric pressure in connection with an orifice system. The buildings in Detroit having orifices in the radiator inlets and using sub-atmospheric pressure have no such difficulty. There are air leaks, of course, but if the air leaks are found and taken care of, there does not appear to be any difficulty with distribution. The heating in general is satisfactory.

Mr. Donnelly mentioned that orifices should be carefully made and free from burrs. One way of getting away from burrs is to use punched stampings. Punched stampings placed in the unions of the radiator valves like gaskets, make a simple and accurate orifice, easy to install. As was suggested, there are times when orifices have to be changed and this can be done easily, if they are installed in a valve, by breaking the union and slipping one out and the other in.

Mr. Donnelly asked about the noise in radiators. We have tried these orifices in both cast-iron and copper radiators. The copper radiator tried was a Murray radiator and strange to say there seemed to be less noise from that radiator than from the cast-iron radiator. That may or may not be true in all cases and in all types of extended surface radiators.

The trouble mentioned about dirt lodging in small orifices can be eliminated by putting a screen ahead of the orifice just as many manufacturers of thermostatic radiator valves now put screens ahead of valves to prevent dirt lodging under the seat.

Dr. Adler spoke of the difficulty of mixture of water and steam in pipe lines and its disturbing influence. Of course, the mains and risers should be well dripped and water should be kept out of the lines.

A high drop through orifices was suggested in the paper to eliminate the use of different sizes of orifices for various distances from the point of control. If a small orifice with a high pressure drop through it is used, thus making the pressure drop through the orifice high in comparison with the drop through the piping, then the pressure drop through the piping becomes an unimportant factor in the distribution.

Mr. West asked about the comparative saving with orifices and with thermostatic radiator valves. The orifice is inexpensive. A stamping with an orifice in it probably costs from 1½ to 10¢, depending upon the size and the quantity and whether a special die must be made.

The cost of the installation is small. In one large Detroit office building the average cost of installing the orifices in the radiator valves was 10¢ per radiator. Probably 25¢ per radiator would be a generous figure for the cost of the orifices installed. Most temperature control systems are much more expensive. In large quantities the average thermostatic radiator valve costs from about $15 up to $30. Several makes are listed at $30 each.

The *National District Heating Association* recently made an investigation of the savings obtained with different types of temperature control which indicated that there was little difference between the economy of heating in buildings with indi-

vidual thermostatic radiator valves and in other buildings equipped with simple radiator orifices. It does not make much difference how the temperature of a building is controlled provided that it is controlled accurately. Many of the less expensive means of controlling temperature are almost as good as those that cost a great deal.

As far as zoning is concerned, I have no accurate figures on that. The data collected by the *National District Heating Association* were not conclusive as regards results of zoning. There undoubtedly is some saving, but it is difficult to obtain comparable data to prove it. It is difficult also to make just comparisons of different kinds of temperature control. There are so many variable factors that affect the figures, that a careful comparison must be made of a large number of cases. The results we have obtained in Detroit indicate that a system with orifices perhaps saves 10, 15, possibly 20 per cent over a system which is controlled by hand. It is possible to get similar results with other types of temperature control.

No. 901

FRICTION HEADS IN ONE-INCH STANDARD CAST-IRON TEES

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This paper is the result of research conducted at Agricultural and Mechanical College of Texas in cooperation with the A. S. H. V. E. Research Laboratory.

THE object of this investigation was to determine the friction heads in a one-inch standard cast-iron tee when 70 F water was flowing through in the several possible directions.

APPARATUS

The apparatus used for the investigation is shown in Figs. 1 and 2. In Fig. 1, *T* represents the tee under investigation and *AT*, *TB* and *TC* represent sections of standard black-iron pipe each about 5 ft long. The three pipe sections were cut from the same piece of pipe in order that the diameters of the three sections might be as nearly equal as possible. *R1*, *R2*, *R3*, *R4* and *R5* represent piezometer rings, to determine the pressure heads at their respective locations. *G1*, *G2*, *G3*, *G4* and *G5* represent glass tubes for obtaining the differences between the pressure heads at the several piezometer rings. *D1* and *D2* represent ½-in. circular diaphragm orifices for determining the quantity of water flowing in the respective pipe sections. *V1* and *V2* represent gate valves for regulating the flow of water through the two respective pipe sections.

During a test, water entered the apparatus at *A* (Fig. 1) and was discharged at *B* and at *C*, the relative quantities flowing in the two directions being regulated so that the quantity flowing out at *B* varied from zero to 100 per cent, while the quantity flowing out at *C* varied from 100 per cent to zero. Provisions were made by which the water flowing through the apparatus was maintained at a mean temperature of 70 F with a maximum variation from that mean of less than one degree.

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Fig. 2 shows a photograph of the apparatus while water was flowing so that 82 per cent was being discharged at *B* and 18 per cent at *C*. For this particular test a small quantity of red dye was added to the water to color it so that the elevations of the water in the glass tubes (*G*) could easily be seen in the photograph.

CALCULATION OF FRICTION AND VELOCITY HEADS

The friction head for this case was calculated as follows: The difference in the elevations of the water in *G1* and *G3* (0.195 in.) was the loss in pressure

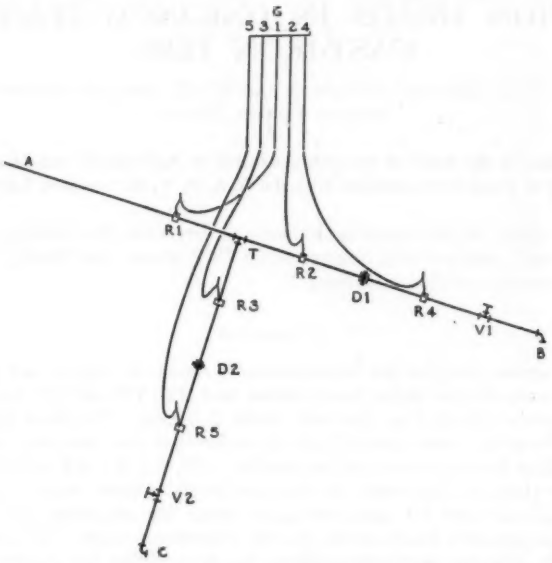

FIG. 1. APPARATUS USED IN DETERMINING FRICTION HEADS IN A 1 x 1 x 1-IN. TEE

head while the water was flowing from *R1* to *R3*. The difference in the elevations of the water in *G3* and *G5* (0.435 in.) was the loss of head while the water was flowing from *R3* to *R5* and through the orifice *D2*. From the calibration of this orifice, referred to in the paper Loss of Head in Submerged Orifices, by F. E. Giesecke,[3] it was found that the velocity of the water in the pipe section *TC* was 3.4 in. per second. However, in order to check this result, the water flowing through the pipe section *TC* was weighed, and the velocity determined from this weight was found to be 3.33 in. per second. In a similar manner, the velocity of the water flowing through the pipe section *TB* was

[3] See A. S. H. V. E. TRANSACTIONS, Vol. 36, 1930.

found to be 15.6 in. per second. The velocity in the pipe section AT was therefore equal to 3.4 in. plus 15.6 in., or 19.0 in. per second.

The velocity heads for the pipe sections AT and TC were then calculated by the formula

$$h = \frac{v^2}{2g}$$

and found to be 0.015 and 0.466 in., respectively. The difference between these two velocity heads (0.451 in.) was the loss in velocity head while the water was flowing from *R1* to *R3*. Adding the loss of velocity head (0.451) to the loss of pressure head (0.195), the total loss of head was found to be 0.646 in. This loss of head may be divided into three parts: The friction head in the

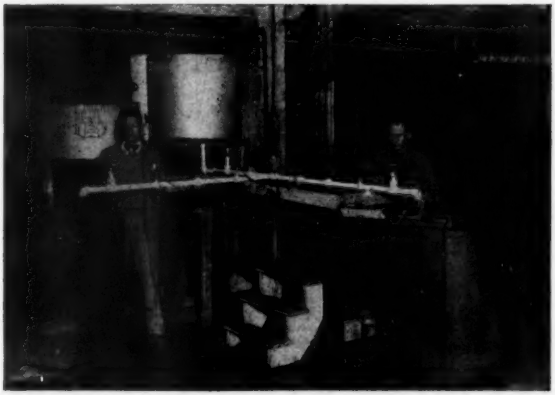

FIG. 2. APPARATUS USED IN DETERMINING FRICTION HEADS IN A 1 x 1 x 1-IN. CAST-IRON TEE

pipe section *R1-T*, the friction head in the tee, and the friction head in the pipe section *T-R3*.

The two pipe friction heads were calculated by the formula

$$h = 0.00685 \frac{v^{1.77}}{d^{1.275}} \text{ (feet of water)}^4$$

and found to be 0.177 in. and 0.008 in., respectively. Subtracting these two friction heads from the total loss of head, the friction head in the tee was found to be 0.461 in. Dividing this by the friction head for a 1-in. standard cast-iron elbow (0.014 in.) calculated by the formula

$$h = 0.0141 \frac{v^{1.96}}{d^{0.26}} \text{ (feet of water)}^4$$

the friction head in the tee was found to be equivalent to 33 elbows. For this case, 18 per cent of the water flowing in the section AT was diverted at right

[4] The Friction of Water in Pipes and Fittings, by F. E. Giesecke (University of Texas Bulletin No. 1759, October 20, 1917).

angles through the tee into the section *TC* and the velocity of the water in this section was 3.4 in. per second. This value (33 elbow equivalents) was recorded in Fig. 3.

By varying the percentage of water flowing through the section *TC*, other values were found and recorded in Fig. 3 and a curve drawn through the points. From this curve it appears that when 100 per cent of the water is flowing through the pipe *TC*, the friction head in the tee is equivalent to about 1.8 elbows and that the number of elbow equivalents increases to about 27 when the proportion of water diverted at right angles is decreased from 100

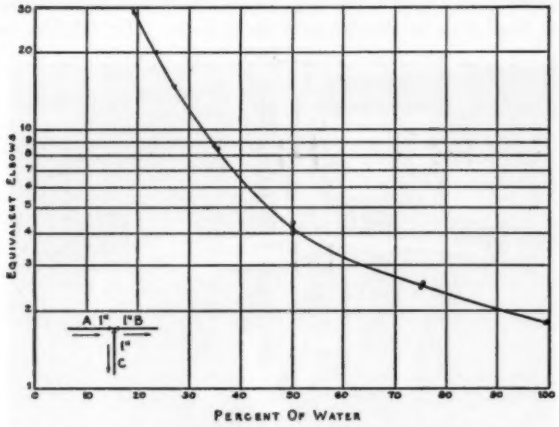

FIG. 3. THE FRICTION HEAD IN A 1 x 1 x 1-IN. CAST-IRON TEE WHEN WATER ENTERS AT *A* AND A PORTION IS DISCHARGED AT *C*, IN TERMS OF THE FRICTION HEAD OF AN ELBOW AT *C*

per cent to 20 per cent. It also appears that the number of eblow equivalents approaches infinity as the percentage of water flowing through the section *TC* approaches zero.

The method just described was also used in finding the friction head for the water flowing straight through the tee. The results for this case were recorded in Fig. 4 and the corresponding curve drawn. It appears from this curve that when 100 per cent of the water is flowing straight through the tee the friction head is equivalent to about 0.06 elbow, and that the elbow equivalents are increased to about 14 when the quantity of water flowing straight through the tee is decreased to 20 per cent of that flowing into the tee. In both cases the elbow equivalent friction head is based on the velocity of the water in the pipe through which the water is discharged from the tee.

CHECK OF RESULTS

Since the results found in this investigation differed materially from those which had been expected, it was decided to check them with the apparatus

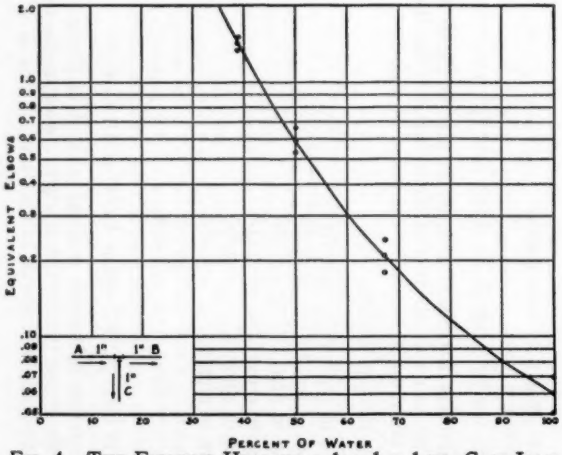

FIG. 4. THE FRICTION HEAD IN A 1 x 1 x 1-IN. CAST-IRON TEE WHEN WATER ENTERS AT *A* AND A PORTION IS DISCHARGED AT *B*, IN TERMS OF THE FRICTION HEAD OF AN ELBOW AT *B*

FIG. 6. APPARATUS USED TO CHECK THE FRICTION HEADS IN TEES AS DETERMINED BY THE APPARATUS SHOWN IN FIGS. 1 AND 2

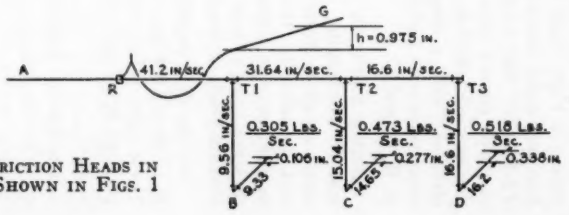

FIG. 5. APPARATUS USED TO CHECK THE FRICTION HEADS IN TEES AS DETERMINED BY THE APPARATUS SHOWN IN FIGS. 1 AND 2

shown in Figs. 5 and 6. In Fig. 5, *T1*, *T2*, and *T3* represent the three 1 x 1 x 1-in. tees under investigation; *A-T1*, a section of pipe 2 ft. long; *T1-T2*, *T2-T3*, *T1-B*, *T2-C*, and *T3-D*, sections of pipe 1 ft. long; *B*, *C*, and *D*, standard 1-in. cast-iron elbows with 1 x 4-in. nipples looking up and reamed to an internal diameter of 1.0625 in.; *R*, a piezometer ring, and *G*, an inclined glass manometer tube with a slope of 1 on 10. The point *O* is the zero point on the manometer tube and is at the same level as the upper ends of the three nipples at *B*, *C*, and *D*.

During a test, the water entered the apparatus at *A* and escaped at *B*, *C*, and *D*, discharging into three separate weighing tanks. The quantities of water escaping from the three nipples were 0.305, 0.473, and 0.518 lb per second, respectively, from which the velocity heads at discharge, the velocities in the six pipe sections, and the velocity head at *R* were calculated. The pressure head (h) at *R* was read in the glass tube. Having these data, the friction heads in the three circuits were calculated.

For the data represented in Fig. 5, the friction heads were as follows: Path *R-T1-B* = 3.088 in.; path *R-T2-C* = 2.913 in.; and path *R-T3-D* = 2.857 in. These three values were determined by deducting the sum of the velocity head, pressure head, and elevation head at the discharge from the sum of those heads at *R*.

The friction head in the path *R-T1-B* includes

Friction in pipe *R-T1*	= 0.700
Friction in pipe *T1-B*	= 0.052
Friction in elbow *B*	= 0.107
Friction in nipple *B*	= 0.017
Total	= 0.876

This leaves for the friction in tee *T1* 3.088 minus 0.876, or 2.212 in.

The friction in tee *T1* is therefore 20.6 times as large as that in elbow *B*. In this case 23.3 per cent of the water is diverted at right angles in the tee so the result checks well with the ratio 20 shown by the curve of Fig. 3.

The friction head in the path *R-T2-C* includes

Friction in pipe *R-T1*	= 0.700
Friction in tee *T1*	= 0.220
Friction in pipe *T1-T2*	= 0.436
Friction in pipe *T2-C*	= 0.117
Friction in elbow *C*	= 0.264
Friction in nipple *C*	= 0.037
Total	= 1.774

This leaves for the friction in tee *T2* 2.913 minus 1.774, or 1.139 in.

The friction in tee *T2* is therefore 4.34 times as large as that in elbow *C*. In this case 48.7 per cent of the water is diverted at right angles in the tee so the result checks well with the ratio 4.76 shown in the curve of Fig. 3.

The friction in the path *R-T3-D* includes

Friction in pipe *R-T1*	= 0.700
Friction in tee *T1*	= 0.220
Friction in pipe *T1-T2*	= 0.436
Friction in tee *T2*	= 0.310
Friction in pipe *T2-T3*	= 0.137
Friction in pipe *T3-D*	= 0.137
Friction in elbow *D*	= 0.310
Friction in nipple *D*	= 0.047
Total	= 2.297

This leaves for the friction in tee *T3* 2.857 minus 2.297, or 0.560 in. The friction in tee *T3* is therefore 1.81 times as large as that in elbow *D*. In this case, 100 per cent of the water is diverted at right angles in the tee so the result checks well with the ratio 1.80 shown in the curve of Fig. 3.

CONCLUSIONS

There is a material variation in the friction heads of tees with the percentage of water deflected through the tee, as determined in this investigation and shown by the curves of Figs. 3 and 4. This variation is of importance in the design of piping installations and particularly when applied to hot water heating systems in which slight differences in available heads may produce serious changes in the operation of the heating system.

It is important that additional investigations be conducted to determine the friction heads in tees when the water flows in other directions than those discussed in this paper, to determine how the friction head varies with the absolute velocity through the tee, and to determine the difference between the friction heads in a 1 x 1 x 1-in. tee and those in a 1 x 1 x ¾-in. tee.

Investigations along these lines are now under way and results will be reported as early as practicable.

DISCUSSION

M. L. ENGER (WRITTEN): The most striking part of this paper, which brings out several facts of importance in the design of piping for hot water heating systems, is probably Fig. 3 showing that the loss of head in flow out of the side of a tee is surprisingly large when expressed in terms of the number of elbows required to produce the same loss of head when only a portion of the flow is diverted.

The following treatment of the loss of head due to flow from the side of a tee explains why the loss is large for small diversions. The conditions of flow are represented in Fig. A. Most of the water passes straight through the tee with little disturbance, but the water which is diverted impinges against the side of the outlet, causing a violent eddy formation in the side outlet. In such a case it is reasonable to assume that the entire velocity head would be lost in impact. As the amount of diversion increases the loss will approach the loss of head which occurs when the entire flow is out of the side of the tee, due to the fact that the angle of impact becomes flatter. This explanation fits the experimental results quite well, as will be shown.

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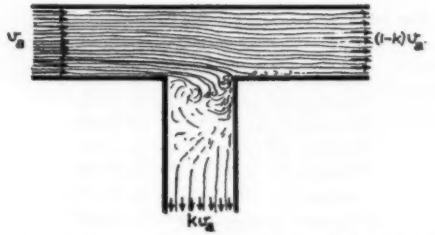

FIG. A. DIAGRAMMATIC SKETCH OF FLOW PARTIALLY DIVERTED IN A 1 X 1 X 1-IN. TEE

The authors' formula for loss of head in a cast-iron elbow is

$$h = 0.0141\frac{v^{1.96}}{d^{0.26}}$$

in which

h = loss of head in feet
v = velocity in feet per second
d = diameter in inches

This may be changed into the following form, with little error, when $d = 1$ in.:

$$h = 0.91\frac{v^2}{2g} \tag{1}$$

The loss of head due to water flowing out of the side of a tee (small diversion), according to the assumption previously given, would be

$$h = \frac{v_a^2}{2g} \tag{2}$$

in which

v_a = velocity of approach

Let k = the fraction of the flow diverted into side opening of tee; then

$$v = kv_a$$

and the loss of head in a standard cast-iron elbow 1 in. in diameter would be

$$h = 0.91\frac{(kv_a)^2}{2g} = 0.91k^2\frac{v_a^2}{2g} \tag{3}$$

Hence the number of 1-in. elbows required to produce the loss of head expressed in Equation 2 would be,

$$n = \frac{v_a^2}{2g} \div 0.91k^2\frac{v_a^2}{2g} = \frac{1}{0.91k^2} \tag{4}$$

In Table A the equivalent number of elbows, n, calculated from Equation 4 are compared with values taken from the authors' Fig. 3.

TABLE A. EQUIVALENT ELBOWS, n

k	Calculated	Authors' Exp.
0.10	111.0	....
0.20	28.0	28.0
0.30	12.0	12.0
0.40	6.9	6.4
0.50	4.4	4.2
0.60	3.1	3.2
0.70	2.2	2.7
0.80	1.7	2.3
0.90	1.4	2.0
1.00	1.1	1.9

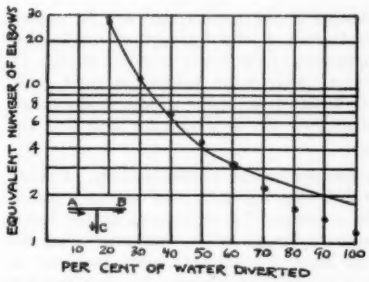

FIG. B. COMPARISON OF THE RESULTS OF GIESECKE AND BADGETT'S EXPERIMENTS WITH VALUES CALCULATED FROM EQUATION (d). THE LINE REPRESENTS THE EXPERIMENTS AND THE POINTS WERE FOUND FROM THE EQUATION

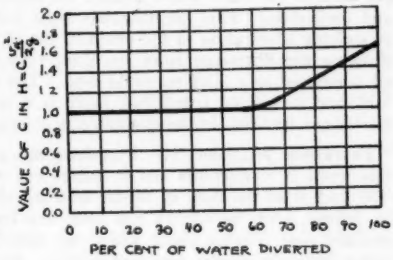

FIG. C. RELATION BETWEEN THE PER CENT OF WATER DIVERTED AND THE CORRECTION FACTOR TO BE USED WITH EQUATION (d) TO OBTAIN THE LOSS OF HEAD IN A 1 x 1 x 1-IN. TEE

The calculated results are plotted in comparison with the authors' curve in Fig. B. It will be noted that the agreement is good for diversions up to 60 per cent and that for greater diversions the calculated results are too low.

From the foregoing it would seem to be simpler to use Equation 2 than to use the equivalent elbows method when the diversion is less than 60 per cent. For greater diversions a correction curve can be used to determine the factors by which the loss of head calculated from Equation 2 should be multiplied. Such a curve is shown in Fig. C.

The second set-up of pipes brought out the interesting results that the discharges in the three branches increased in the downstream direction. At first sight, this seems to be unreasonable because it would seem that the additional friction in the longer paths of flow would produce the opposite effect. However, when the velocity heads are included in the Bernoulli equation it is seen that the losses of head are fully accounted for. This phenomenon must be taken into account in many cases in design. For example, in the wash water system in water filters if high velocities

FIG. D. EFFECT OF MOMENTUM OF WATER

are used in the wash water piping, the distribution of the wash water will be uneven and ineffective. This question has been investigated at the University of Illinois.[1] A striking illustration of the effect of the momentum of the water is given in Fig. D, showing the discharge from ten 2-in. riser pipes with ¾-in. orifices at their ends, all connected to a 2-in. manifold pipe. The flow was from right to left in the manifold, and it will be noted that there is a small flow in the first riser and that the discharges increase in the downstream risers.

PROFESSOR WILLARD: Mr. President, my comments are all commendatory. I think this work is worthwhile and it brings out a fact that has interested me very much. The fact is in the flow of fluids, whether air or steam, the resistances or the losses in fittings, such as elbows and tees, are frequently much more important than the straight pipe friction about which we seem to know a good deal more than we do about losses in elbows, tees and fittings. Dr. Giesecke has brought this fact forcibly to our attention in connection with the flow of fluids in hot water heating systems at low velocities. If the resistance in such fittings is bad at low velocities, it is going to be much more so at high velocities, and more work along this line would be desirable from the standpoint of the Society and also for the benefit of the heating and ventilating engineers of this country.

PRESIDENT CARRIER: As Professor Willard stated, this is of interest to every one having to do with the flow of fluids. In the handling of air, we have a similar problem. Fortunately, however, we are not confined to such conduits so barbarous from a mechanical standpoint as a cast-iron tee. We can make the tee connections so that the friction is considerably reduced by the use of a splitter. In fact, with extreme care, where it is necessary, the losses can be reduced to small amounts. We very seldom have any abrupt right-angle connections in the flow of air. Sometimes that is found in small sizes of piping at low velocities and high static heads.

[1] Pressures in Manifold Pipes, *Journal American Water Works Association*, Vol. 21, No. 5, May, 1929.

No. 902

SIZING PIPES AND ORIFICES FOR GRAVITY HOT WATER HEATING SYSTEMS

This paper is the result of research conducted at the Engineering Experiment Station of Texas Agricultural and Mechanical College in cooperation with the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS.

By ELMER G. SMITH,[1] COLLEGE STATION, TEXAS
MEMBER

A SET of tables intended for use in designing gravity circulation hot water heating systems for residences and other buildings of small or moderate size, was presented by the writer at the 36th Annual Meeting of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, in Philadelphia, Pa., January, 1930.[2] After a careful analysis of these tables, the Technical Advisory Committee on Pipe Sizes for Heating Systems (1930-31) came to the conclusion that the pipe sizes obtained by means of them would result in systems too costly for general use. It was therefore decided to continue the investigation in order to develop another set of tables by which smaller and more economical pipe sizes could be obtained. It also seemed desirable to revise the orifice formula so that it would give more accurate results for small "radiators." In order to reduce pipe sizes without reducing radiator efficiencies it was necessary to reduce safety factors, and to reduce safety factors without jeopardizing circulation required much more accurate calculations than those upon which the first tables were based.

Every effort has been put forth to make the tables presented in this paper as practical as possible. The present study covers bare mains. A study of insulated mains has not yet been made, but it seems probable that these data will give satisfactory results if the insulation has a low efficiency. It is not necessarily true that a system having covered mains is better than a system having bare or partly covered mains. The bare mains not only contribute considerable heat to the basement but also by warming the basement ceiling they help to heat the first story and make it more comfortable.

[1] Assistant Professor of Physics, Texas Agricultural and Mechanical College.
[2] See Pipe and Orifice Sizes for Small Gravity Circulation Hot Water Heating Systems, by Elmer G. Smith, A. S. H. V. E. TRANSACTIONS, Vol. 36, 1930.
Presented at the Semi-Annual Meeting of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, Swampscott, Mass., June 1931.

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Where basement "radiation" is required it is essential that a ceiling type of "radiator" be used and connected in a definite manner as explained later in this paper.

DEFINITIONS

In order to avoid confusion, the following terms which are used in this paper, have the definitions indicated:

Riser. A vertical pipe.
Boiler Uptake. A vertical pipe leading up from a boiler outlet to a supply main or header.
Boiler Return. A vertical pipe leading down from a return main to a boiler inlet or header.
Supply Mains. The horizontal piping leading from the boiler uptake to the supply "radiator" or riser branches.
Return Mains. The horizontal piping leading from the return "radiator" or riser branches to the boiler returns.
Supply Riser Branch. The piping which connects a supply main to a supply riser.
Return Riser Branch. The piping which connects a return riser to a return main.
Supply "Radiator" Branch. The piping which connects a supply main or riser to a "radiator."
Return "Radiator" Branch. The piping which connects a "radiator" to a return main or riser.
Expansion Pipe. The piping which connects the system to the expansion tank.
Overflow. The piping which connects the top of the expansion tank to the sewer or other means of drainage.
Circulating Pipe. The piping which connects the expansion tank to the bottom of the boiler or to some part of the return piping.
Vent Pipe. The pipe at the top of the overflow which is open to the atmosphere in order to prevent syphoning the system.

Although the tables contained in this paper were calculated primarily for use in connection with systems having open expansion tanks, they can also be used for closed systems. An open expansion tank system designed according to these tables and operated at its maximum capacity should show an average temperature differential of about 30 deg for the "radiators," being higher than this for "radiators" close to the boiler and lower for "radiators" far from the boiler. Under these conditions the temperature differential at the boiler should be about 40 deg in an average installation. When a system is being operated at less than its maximum capacity the temperature differentials will all be lower than the calculated values but their relative values will be about the same. On the other hand, if a system is put under pressure and made to operate at a capacity greater than would be possible for a similar open system, the temperature differentials will all be somewhat higher than for the open system.

From this it can be seen that if the pipe and orifice sizes for a closed system are selected on a basis of the "radiation" actually used, the system will operate at a higher temperature differential than would an open system, and the blow-off valve will have to be set to maintain a pressure of at least 15 lb per square inch at the top of the highest "radiator" when the system is being operated at its maximum capacity.

If, on the other hand, the pipe and orifice sizes are selected as if for an open system and only the sizes of the "radiators" reduced, the system will operate at about the same temperature differential as an open system. In this case the blow-off valve can be set to maintain a pressure of 10 lb per square inch at the top of the highest "radiator" when the system is being operated at its maximum capacity.

The first method makes the cost of installation less and gives smaller heat losses in the basement, which makes it more suitable for a system having bare mains. However, the second method is less likely to cause trouble by leaking because of the lower pressure and lower maximum temperature of the supply piping.

RESULTS OF TESTS ON A THREE-STORY EXPERIMENTAL PLANT

The platforms and "radiators" used to test these tables were the same ones that were used to test the tables submitted in the original paper on this subject.[2] The converter used to heat the water for the system was altered in order to reduce its internal resistance. Even after the alterations, the internal resistance of this converter may have been higher than that of an ordinary "boiler" having the same size outlets and inlets, but the difference was not enough to be serious. The sizes of the pipes and orifices were changed to correspond with those given in the new tables. The layout of the experimental plant is shown in Fig. 1. "Radiators" No. 16, No. 26, and No. 36 are not a part of the system as originally designed and were used principally to determine the effect of overloading and to permit one large "radiator" (No. 16) to be substituted for the three small ones on the end of the line (No. 15, No. 25, and No. 35). There was no insulation on any part of the mains, risers, riser branches, or "radiator" branches during any of these tests.

The first test on this system was run with the "radiators" standing on the scaffold with both front and back exposed and the mains running less than one foot above a cold floor. There was no flooring or ceiling above the mains such as there would have been in an ordinary building. Consequently, the cooling of the water in the mains was greater than it would have been under average conditions. In the remaining tests the "radiators" and risers were backed on one side by wall board (See Fig. 2) in order to approximate the effects of a wall, and the mains were partly covered by wall board as illustrated in Fig. 3. The wall board was applied loosely with numerous holes around risers and scaffolding, and one end was left open so that the enclosure would not become too hot. This may not have closely simulated average basement conditions but it at least provided a warm ceiling with a cool floor and only a moderate amount of convection. The results of these tests are given in Table 1.

In the first test, the "radiators" and pipes were exposed much more than they would have been in an ordinary building. The hottest "radiator" was No. 23 with an average temperature of 167½ F, and the coolest "radiators" were Nos. 25 and 35, each of which had an average temperature of 155 F, a difference of 12½ deg. However, the heat dissipated per square foot of equivalent "radiation" was practically the same for all three of these "radiators" because No. 25 and No. 35 were each only five-section "radiators" while No.

[2] See Pipe and Orifice Sizes for Small Gravity Circulation Hot Water Heating Systems, by Elmer G. Smith, A. S. H. V. E. TRANSACTIONS, Vol. 36, 1930.

TABLE 1. RESULTS OF TESTS ON EXPERIMENTAL PLANT

First Test: No basement ceiling or walls. Expansion tank open. Readings taken 3 hours after starting.

Radiator Number	Boiler	11	12	13	14	15	21	22	23	24	25	31	32	33	34	35
Supply Temperature	190½	186	182	176	179	170½	184½	180½	181	178	167	181	177	179	169	163
Return Temperature	144	145	149	145	152½	147	150	146	154	155	143	141	143½	151	150½	147
Average Temperature	167¾	165½	165½	160½	165¾	158¾	167¼	163¼	167½	166½	155	161	160¼	165	159¾	155
Temperature Differential ...	56½	41	33	32	26½	23½	34½	34½	27	23	24	40	33½	28	18½	16

Air Temperatures: First floor 59 F, Second floor 60.5 F, Third floor about 61 F.

Second Test: Basement ceiling and walls simulated. Expansion tank open. Readings taken 2½ hours after starting.

Radiator Number	Boiler	11	12	13	14	15	21	22	23	24	25	31	32	33	34	35
Supply Temperature	188	186	182	178	179	172	185½	181	182	179½	169	181	177	179	170	164
Return Temperature	146	146	152	152	156	152	153	151	158	159	151	145	140	153	159	149
Average Temperature	167	166	167	165	167½	162	169¼	166	170	169¼	160	163	158½	166	164½	156½
Temperature Differential ...	42	40	30	26	23	20	32½	30	24	20½	18	36	37	26	11	15

Air Temperatures: First floor 71 F, Second floor 69.5 F.

Third Test: Basement ceiling and walls simulated. Expansion tank closed. Maximum pressure 30 lb per square inch at gage, or 20 lb per square inch at the expansion tank. Readings taken 3½ hours after starting.

Radiator Number	Boiler	11	12	13	14	15	21	22	23	24	25	31	32	33	34	35
Supply Temperature	256	249	245	240	242½	234	247	244	245	242	229	245	243	242½	231	223
Return Temperature	201½	194	204½	208	216	208½	198½	203	215	215	202	211½	207	212	213	204½
Average Temperature	228¾	221½	224¾	224	229¼	221¼	222¾	222	230	228½	216½	228¼	225	227½	222	213¾
Temperature Differential ...	55½	55	41½	32	26½	26	48½	41	30	27	27	34½	36	30½	18	18½

Air Temperatures: Basement 79 F, First floor 69.5 F, Second floor 70.5 F, Third floor 76 F.

Fourth Test: Basement ceiling and walls simulated. Expansion tank open. Readings taken 3¼ hours after starting.

Radiator Number	Boiler	11	12	13	14	16	21	22	23	24	31	32	33	34
Supply Temperature	204	200	197	192½	193½	189	199	194½	194½	192	196	193	193½	186
Return Temperature	161½	161	168	164	171	163½	166	161	170½	171	166	166	166½	168
Average Temperature	182¾	180½	182½	178¼	182¼	176¼	182½	177¾	182½	181½	181	179½	180	177
Temperature Differential ..	42½	39	29	28½	22½	25½	33	33½	24	21	30	27	27	18

Air Temperatures: Basement 86 F, First floor 84 F, Second floor 82 F, Third floor 86 F.

Fifth Test: A continuation of fourth test. Expansion tank closed. Maximum pressure 25 lb per square inch at the gage, or 15 lb per square inch at the expansion tank. Readings taken 3¼ hours after closing expansion tank.

Radiator Number	Boiler	11	12	13	14	16	21	22	23	24	31	32	33	34
Supply Temperature	251½	244	241	236	238	233	243	239	239	237	238	237	236	227
Return Temperature	199	191	198	203	212	202	197	196	209½	211	207½	204	206	209½
Average Temperature	225¼	217½	219½	219½	225	217½	220	217½	224¼	224	222¾	220½	221	217¼
Temperature Differential ..	52½	53	43	33	26	31	46	43	29½	26	31½	33	30	17½

Air Temperatures: Basement 84 F, First floor 77 F, Second floor 76.5 F, Third floor 83.5 F.

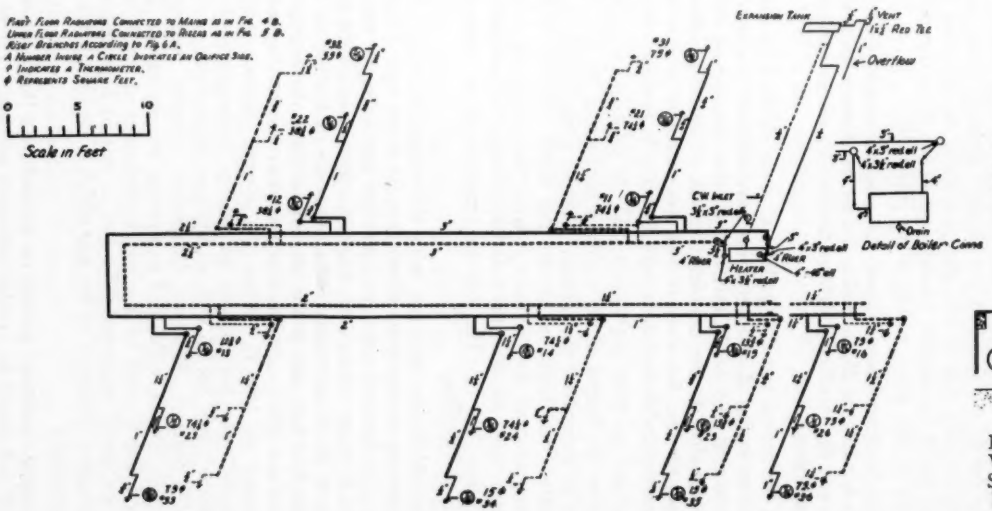

FIG. 1. LAYOUT OF EXPERIMENTAL PLANT

FIG. 2. SUPPLY END OF RADIATOR SHOWING WALL BOARD BACKING

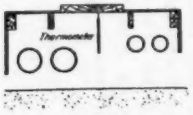

FIG. 3. SECTIONAL VIEW OF MAINS SHOWING WALL BOARD ENCLOSURE

23 was a 27-section "radiator." The "boiler" was intended to operate on a 40-deg differential and on the first floor the temperature differentials were intended to be about 37½ deg in section 1, about 34½ deg in section 2, about 29½ deg in section 3, and about 22½ deg in section 4. Also, the "radiator" temperature differentials were intended to decrease as the heights of the "radiators" above the "boiler" increased. Most of the observed temperature differentials were reasonably close to their calculated values, but the temperature differential of the "boiler" was considerably too large. This may have been partly due to the fact that the "boiler" resistance was a little too high but it seems probable that it was due principally to excessive heat losses from the mains which are usually located just under the cellar ceiling where it is hot, instead of just above a floor where it is cold.

The "radiator" temperature differentials ranged from 41 deg for No. 11 to 16 deg for No. 35, becoming less as the distance from the "boiler" increased. This was the result of a deliberate effort to secure uniform average "radiator" temperatures.

In the second test the "radiators" were shielded on one side by wall board as shown in Fig. 2, and the mains were shielded in a manner somewhat similar as shown in Fig. 3. In this test, all the temperature differentials were satisfactorily close to the calculated values. The temperature differential of No. 34 was too low, but since it was approximately correct in all the other tests, it seems probable that there was an error in reading one of the thermometers.

In both the first and second tests, it required about 10 min for the water to travel the length of the mains, a distance of 98 ft. This corresponds to a speed of about 10 ft per minute. Sixteen minutes from the starting time every "radiator" had begun to receive hot water. Twenty-eight minutes from the starting time all the "radiators" were filled with hot water and by forty-five minutes from the starting time the average temperature of every "radiator" was above 140 F. The average temperatures of some of them were above 150 F at this time.

The third test was run with the expansion tank closed and the steam pressure on the converter raised from 18 lb per square inch to 65 lb per square inch. No blow-off valve was provided but whenever the water pressure went above 30 lb per square inch, gage, the drain was opened until this pressure had fallen to 28 lb per square inch. The rate of heating up was more rapid than in the first test and at the end of an hour the average temperature of the system was 200 F. At this time the water was leaving the heater at 223 F, and returning at 169 F. When the system reached equilibrium, the hottest "radiator" was No. 23 with an average temperature of 230 F. The coolest "radiator" was No. 35 with an average temperature of 213¾ F, a difference of 16¼ degrees. However, the coolest "radiator" having more than five sections was No. 11, which had an average temperature of 221½ F, so that when a correction is made for the higher efficiencies of small "radiators," it is found that the actual maximum difference in rate of heat emission corresponds to a temperature difference of 8½ deg which is very small for such high "radiator" temperatures.

A small "radiator" located on the end of a line is always likely to circulate poorly and should be avoided whenever possible. The experimental plant was

erected in this manner in order to make certain that the tables would give reasonably good results even when the arrangement of the "radiators" was such as to make good circulation difficult.

In order to find out what might be expected from a more normal installation the risers carrying "radiators" No. 15, No. 25 and No. 35 were cut out and "radiator" No. 16 was connected in their place. Since the total rating of the disconnected "radiators" was only 42½ sq ft, while the rating of No. 16 was 75 sq ft, this constituted a slight overload upon the mains, but since the overload amounted to less than half the rating of the last first-floor "radiator" (No. 16), no serious consequences were anticipated. The fourth test was then run with this set-up. When equilibrium had been established, the hottest "radiators" were No. 12, No. 21, and No. 23, each of which had an average temperature of 182¼ F. The coolest was No. 16 with an average temperature of 176¼ F, a difference of 6 deg. Apparently the only effect of the overload was to increase the temperature differential of "radiator" No. 16 from the calculated 22½ deg to 25½ deg and thereby to lower its average temperature by 1½ deg.

As soon as the fourth test was completed the fifth test was started by simply closing the expansion tank vent and raising the steam pressure on the converter to 65 lb per square inch. When the system had again come to equilibrium, it was found that the hottest "radiator" was No. 14 with an average temperature of 225 F. The coolest were Nos. 11, 16 and 22, each of which had an average temperature of 217½ F. The temperature difference between the hottest "radiator" and the coolest was therefore 7½ deg. During this test the pressure at the gage was never higher than 25 lb per square inch, which means that the pressure at the expansion tank never went above 15 lb per square inch.

TYPES OF PIPING TO WHICH TABLES APPLY

The types of first floor "radiator" branches to which these tables are intended to apply are illustrated in Figs. 4*A* to 4*L*, inclusive. The types of branch shown in Figs. 4*A*, *D*, *G* and *J* probably give somewhat better circulation through the radiator than any of the others and those shown in Figs. 4*B*, *E*, *H* and *K* probably give slightly better "radiator" circulation (higher average radiator temperature) than those shown at *C*, *F*, *I* and *L*. Also, the high supply connection places the valve in a convenient position where it is not necessary to stoop to operate it. On the other hand, the low supply connection takes longer to become air bound than the high supply connection.

Two possible methods of connecting basement "radiators" are shown at Figs. 4*M* and 4*N*. These connections contain more elbows than are necessary but if fewer elbows were to be used for basement "radiators" than for first floor "radiators," it would complicate the tables and this did not seem to be advisable. For basement "radiators" it is always advisable to keep the supply radiator branch above the return radiator branch. The connections shown at Fig. 4*M* are better from the standpoint of circulation because the supply branch is higher and shorter, but the valve is easier to reach in Fig. 4*N*. The connections to the mains are usually made as illustrated in Figs. 4*A*, *B* and *C*, but the method shown in Figs. 4*D*, *E* and *F* is just as good except that the return branch is not quite as flexible.

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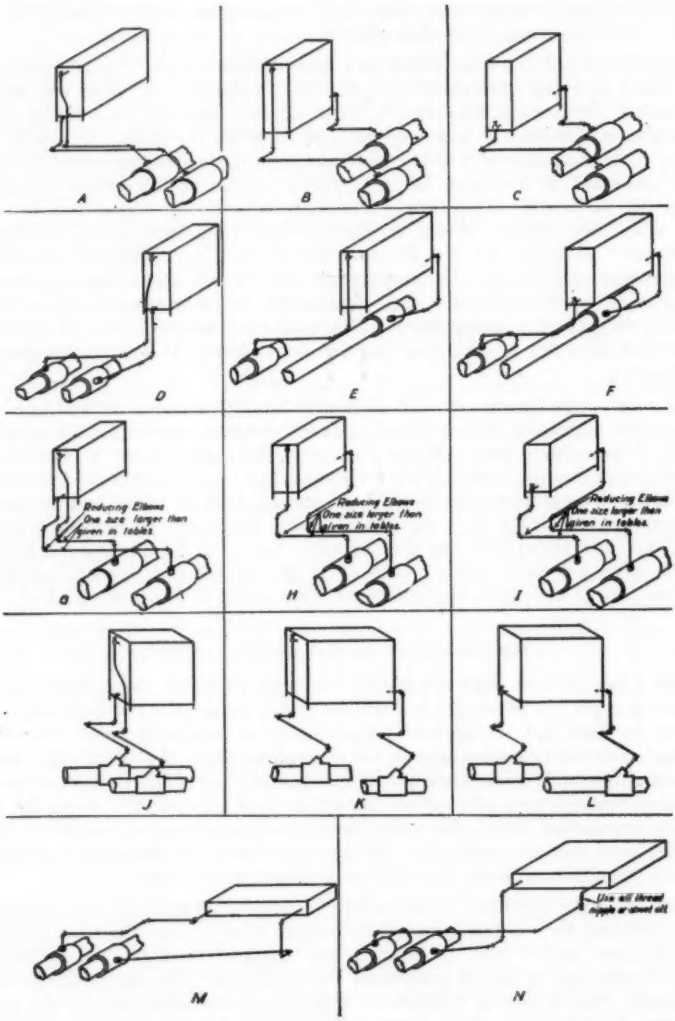

FIGS. 4A TO 4N. FIRST-FLOOR RADIATOR CONNECTIONS TO WHICH TABLES ARE INTENDED TO APPLY

In using the connections shown in Figs. 4*D*, *E* and *F*, it is important to note that the supply main must be farther from the "radiator" than the return main in order that the connection to the supply main may be vertical and to the return main horizontal. Close to the "boiler," this precaution may be disregarded, but in connecting "radiators" near the ends of the mains, it is important. The connections shown in Figs. 4*G*, *H* and *I* are extremely flexible and should be used whenever the "radiator" riser passes through a tight slot or hole that will not permit it to move freely. This is especially important near an elbow in the mains because at such places the mains move toward and away from the wall as well as parallel to it. The connections shown in Figs. 4*J*, *K*, *L* give better circulation through the mains than any of the others because the velocity heads are not entirely lost as the water enters and emerges from the "radiator" branches, but the large *Y* connections may be difficult to obtain and the connection as a whole is not quite as flexible as some of the others.

It is expected that the length of a "radiator" branch, measured horizontally from the mains to the "radiator" will normally be about 4 ft. If this distance is more than 8 ft it is advisable to use the orifice and pipe sizes that are ordinarily required for the next larger "radiator."

The methods of connecting the "radiators" on floors higher than the first are shown in Figs. 5*A*, 5*B* and 5*C*. Any other type of connection containing approximately the same number of elbows and having the same vertical flexibility is equally satisfactory. It is expected that on the average a "radiator" will be about 4 ft from its risers. As in the case of the first-floor "radiators," it is good practice to use the orifice and pipe size ordinarily required for the next larger "radiator" when the "radiator" is more than 8 ft from the risers.

It is usually advisable to anchor or support a riser at the point where the second floor "radiator" branches enter it. This allows for a considerable amount of horizontal flexibility at the base of the riser where it joins the riser branch and at the same time prevents any appreciable vertical motion at this point.

The riser branches should be installed according to Figs. 6*A*, 6*B*, 6*C* or 6*D*. When the riser branch is larger than the riser the change should be made by means of a reducing elbow at the foot of the riser. That is, all the fittings must be in the branch and none in the riser. This is important.

The mains may be run either side by side as in Fig. 7*A*, or the supply main may be placed directly above the return main, as in Fig. 7*B*. The side-by-side arrangement gives more head room but the supply-main-above arrangement makes it easier to connect branches and also insures somewhat more stable circulation. Either arrangement, however, is entirely satisfactory, provided the remainder of the installation is properly made. The return main should never be run above the supply main as this might cause instability.

In calculating the capacities of the mains, allowance was made for four elbows in each main. This number should not be exceeded unless compensated by the use of larger pipe. Elbows located so near the end of the line that they affect only one "radiator" can usually be neglected, but elbows near the "boiler" are very important and should be avoided whenever possible. If it is necessary to locate more than one elbow in each main near the "boiler," the best plan is to use larger fittings. One size larger will permit two elbows instead of one and two sizes larger will permit four. It should be remembered

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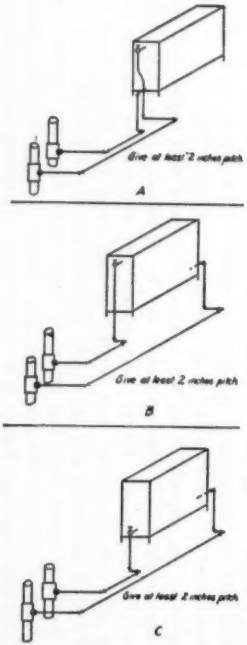
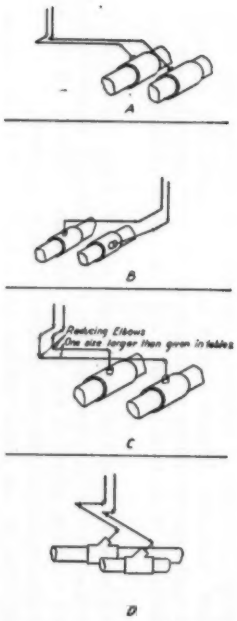
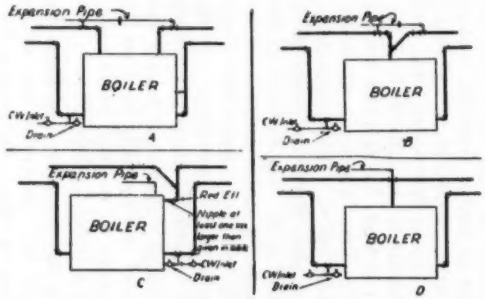


FIGS. 5A TO 5C. METHODS OF CONNECTING RADIATORS ABOVE FIRST FLOOR

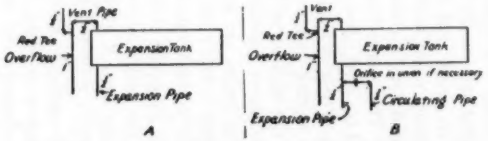


FIGS. 6A TO 6D. METHODS OF RUNNING LATERALS

FIGS. 7A AND 7B. ARRANGEMENT OF SUPPLY AND RETURN MAINS



FIGS. 8A TO 8D—METHODS OF CONNECTING THE "BOILER" UPTAKES



FIGS. 9A AND 9B. EXPANSION TANK CONNECTIONS

that a tee is equivalent to 2.2 elbows. Tees should therefore be avoided whenever possible. The elbows or reducing elbows at the tops of the "boiler" uptake and "boiler" return are included in the "boiler" connections and need not be considered when counting the elbows in the mains.

The best method of connecting the "boiler" uptakes and returns is shown at Fig. 8*A*. The connections shown in Figs. 8*B* and 8*C* also give good results. The connections shown at Fig. 8*D* are acceptable provided the line contains not more than three elbows per main, none of which should be close to the heater. Since the mains are usually smaller than the "boiler" uptakes, the elbows at the tops of the "boiler" uptakes and returns will usually be reducing elbows. There is, of course, considerable latitude in the location of the cold-water feed and the drain but there is probably a little less danger of air binding

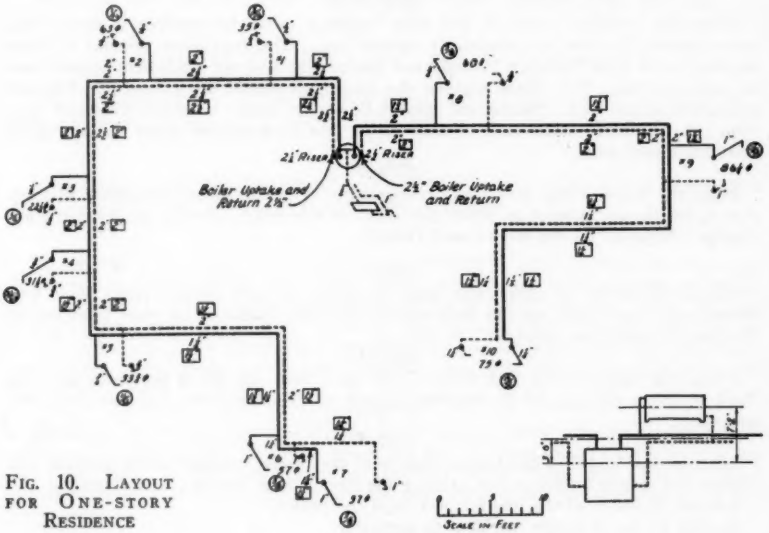


FIG. 10. LAYOUT FOR ONE-STORY RESIDENCE

if the cold-water feed as well as the drain is located near the bottom of the "boiler" because the air driven out of the water by the heat will have more time to collect in large bubbles before leaving the "boiler" and will therefore be more likely to be caught by the expansion pipe and kept out of the "radiators."

The expansion pipe connections shown in Figs. 8*A*, 8*B*, 8*C* and 8*D* are good because they are located so as to divert from the "radiators" a large portion of the air that is driven out of the water in the boiler. Frequently the expansion tank is located in such a cold place that it must be protected from freezing by circulation and in this case a return line or circulating pipe from the expansion tank, preferably to the bottom of the "boiler," is required.

If the expansion tank is located at the top of a riser and connected to both the supply and return it should be treated as a "radiator" and orificed. The

risers should be sized to carry their load of "radiators" plus another "radiator" as large as the expansion tank orifice would carry. If this is not done the pressure difference between the risers may be so reduced that poor circulation will result. The expansion tank connections shown in Figs. 9*A* and 9*B* are good because if the overflow is nearly straight and vertical the reducing tee prevents any possibility of overflow even when the vent pipe is very short.

GENERAL PROCEDURE IN DESIGNING A SYSTEM

For convenience the process of designing a gravity hot water system may be divided into ten steps, as follows:

Step 1.

Select the "radiator" sizes so that each "radiator" shall be required to deliver the same number of Btu per equivalent square foot. The maximum amount of heat required from each "radiator" should not exceed 150 Btu per equivalent square foot for open systems. For closed systems the maximum should not exceed 216 Btu per equivalent square foot. "Radiators" should be located under windows whenever possible. A "radiator" under each window is the best arrangement when comfort takes precedence over cost.

Step 2.

Make the layout using only such arrangements of piping as are illustrated in Figs. 4 to 9, but do not attempt to select pipe sizes at this point. As far as possible place a large "radiator" on the end of each line.

Step 3.

Check the layout to make sure that the mains do not contain more than four elbows each, neglecting the one between the last two "radiator" or riser branches on the line, if such a one occurs.

Step 4.

Check the layout to see that there is not more than one elbow per main near the "boiler" unless the plan of the building is such as to make more than one absolutely necessary.

Step 5.

Measure the length of the longest line from the "boiler" uptake to the farthest end. Divide this length by seven and arrange the line in four sections, as follows:

Section 4: the $\frac{1}{7}$ of the line farthest from the heater.
Section 3: the $\frac{2}{7}$ of the line next to section 4.
Section 2: the $\frac{2}{7}$ of the line next to section 3.
Section 1: the remaining $\frac{2}{7}$ of the line lying between section 2 and the "boiler."

Step 6.

Select the proper sizes of "radiator" branches and orifices from Table 2. Include the expansion tank unless it is separately connected to the "boiler." The height above the mid-point of the "boiler" in feet is the vertical distance in feet from the average height of the "boiler" openings to the average height of the "radiator" openings. Thus, the mid-point of the "boiler" is usually considerably above the actual middle of the "boiler," and the measurement should be made either to the middle or the bottom of the "radiator," depending on whether the high or low supply connection is used. It will usually be found most convenient to begin with the "radiators" in section 4.

Step 7.

If the installation is more than one story high, select the proper sizes for the risers from Table 3 or Table 4 depending on the capacity required. Include the expansion tank unless it is separately connected to the "boiler."

Step 8.

Select the sizes for the mains by means of Table 5 using either the simple or the more accurate method. The *Distance from Mid-point of "Boiler" to Lowest Point of Supply Main* is the vertical distance in feet from the average height of the "boiler" openings to the middle of the supply main at the junction of this main with the "boiler" uptake. The return main may be either on the same level with the supply main or below it, *but the return main must never be above the supply main.*

Step 9.

If there is more than one elbow per main close to the "boiler," increase the size of this portion of the mains as follows:

a. For two elbows per main use one size larger than given in tables.

b. For three or four elbows per main use two sizes larger than given in the tables.

Step 10.

Select the sizes for the "boiler" uptakes and "boiler" returns from Table 6.

SELECTING PIPE SIZES FOR A ONE-STORY RESIDENCE

Fig. 10 represents a layout that has been made in accordance with the foregoing general procedure, as follows:

Step 1.

It is assumed that the "radiators" have been selected and located as specified in step 1 of the *General Procedure in Designing a System.*

Step 2.

The piping is to correspond with the most common practice. Thus, the mains are to be side by side as in Fig. 7*A*. The "boiler" uptakes are to be separate as in Fig. 8*A*. If all the risers pass through reasonably large holes and are free to move in any horizontal direction, all the "radiators" may be connected according to Figs. 4*A* or 4*B*, but if No. 2, No. 5 or No. 9 has risers passing through snug holes or sleeves, each of these "radiators" having such relatively immobile risers should be connected according to Figs. 4*G* or 4*H*.

Step 3.

Checking the layout according to step 3, it will be found that the longer branch contains five elbows per main, but since the last of these occurs between the last two "radiators" on the line it may be neglected, which leaves four, the maximum number allowable. The right hand branch has four elbows per main which is also satisfactory.

Step 4.

Checking again according to step 4, it will be found that there is only one elbow per main close to the heater which, again, is as it should be.

Step 5.

The length of the left hand branch is 8 ft + 23 ft + 23½ ft + 18 ft + 13 ft + 7 ft or 92½ ft. $\frac{92.5}{7} = 13.2$ ft. Therefore, section 4 consists of the 13.2 ft at the end farthest from the "boiler," section 3 consists of the next 26.4 ft, section 2 consists of the next 26.4 ft, and section 1 consists of the remaining 26.4 ft between section 2 and the "boiler."

Step 6.

According to step 5, "radiators" No. 7 and No. 6 lie in section 4 and since the average height of the inlet and outlet is 7 ft. 2 in. above the mid-point of the "boiler," the column headed ⅞ in Table 2, *Section* 4, must be used. Since the values given in this table are maximums, the first number larger than 57 must be selected. This

TABLE 2. CAPACITIES OF ORIFICES AND RADIATOR BRANCHES. IF THE RADIATOR THE MAINS OR RISERS TO THE RADIATOR, USE AN

PIPE SIZE	DRILL SIZE	SECTION 1 — CONNECT TO MAINS ACCORDING TO FIGS. 4A–4N.												HEIGHT ABOVE MIDPOINT			
		3-3½	3½-4	4-4½	4½-5	5-6	6-7	7-8	8-9	9-10	10-11	11-12	12-13	13-14	14-15	15-16	16-17
½	7/64		2.1	2.3	2.4	2.7	2.9	3.1	3.3	3.6	3.7	3.9	4.0	3.4	3.6	3.7	3.9
	33	2.1	2.3	2.5	2.6	2.9	3.2	3.4	3.6	3.8	4.0	4.2	4.4	3.7	3.9	4.1	4.2
	32	2.4	2.6	2.8	2.9	3.3	3.6	3.8	4.1	4.4	4.6	4.8	4.9	4.2	4.4	4.5	4.7
	31	2.7	3.0	3.2	3.4	3.7	4.0	4.3	4.6	4.9	5.1	5.4	5.6	4.6	5.0	5.2	5.4
	1/8	3.0	3.2	3.4	3.7	4.0	4.4	4.7	5.0	5.4	5.6	5.9	6.1	5.2	5.4	5.6	5.8
	30	3.5	3.8	4.1	4.3	4.8	5.2	5.6	6.0	6.4	6.7	7.0	7.2	6.2	6.4	6.7	6.9
	29	3.9	4.2	4.7	4.8	5.3	5.8	6.2	6.6	7.0	7.3	7.7	8.0	6.8	7.1	7.4	7.6
	9/64	4.4	4.8	5.1	5.5	6.0	6.6	7.1	7.5	8.0	8.3	8.7	9.1	7.7	8.1	8.4	8.7
	26	4.9	5.3	5.5	6.0	6.6	7.2	7.7	8.3	8.7	9.2	9.6	10.0	8.5	8.9	9.2	9.6
	24	5.3	5.7	6.1	6.5	7.2	7.8	8.4	8.9	9.4	9.9	10.3	10.7	9.2	9.6	10.0	10.3
	5/32	5.7	6.2	6.6	7.0	7.8	8.5	9.1	9.7	10.2	10.7	11.2	11.7	10.0	10.4	10.8	11.2
	20	6.2	6.7	7.2	7.6	8.4	9.2	9.8	10.5	11.1	11.6	12.2	12.7	10.8	11.2	11.7	12.2
	19	6.8	7.4	7.9	8.4	9.3	10.1	10.8	11.5	12.2	12.7	13.3	13.8	11.9	12.4	12.9	13.4
	11/64	7.7	8.3	8.7	9.4	10.4	11.3	12.1	12.9	13.6	14.3	14.9	15.5	13.4	13.9	14.5	15.0
	15	8.5	9.2	9.9	10.9	11.5	12.5	13.4	14.3	15.2	15.8	16.5	17.2	14.8	15.4	16.1	16.6
	3/16	9.5	10.3	10.8	11.6	12.9	14.0	14.9	15.9	16.9	17.6	18.4	19.1	16.5	17.2	17.9	18.6
	9	10.3	11.2	11.9	12.7	14.0	15.2	16.2	17.3	18.3	19.1	19.9	20.7	18.0	18.8	19.6	20.3
	13/64	11.0	12.0	12.6	13.5	14.9	16.2	17.3	18.5	19.5	20.3	21.2	22.0	19.2	20.0	20.8	21.6
	4	12.2	13.3	13.9	14.9	16.5	17.9	19.1	20.4	21.5	22.4	23.4	24.2	21.3	22.2	23.1	23.9
	7/32	14.2	15.4	16.4	17.4	19.1	20.6	22.1	23.4	24.7	25.8	26.8	27.8	24.7	25.8	26.8	27.9
	15/64	16.1	17.5	18.6	19.7	21.7	23.4	25.0	26.4	27.8	29.0	30.0	31.1	28.0	29.2	30.4	31.6
	1/4	18.1	19.6	20.6	22.0	24.1	26.0	27.7	29.2	30.7	32.0	33.2	34.2	31.2	32.6	34.0	35.3
	17/64	20.0	21.6	22.7	24.2	26.5	28.6	30.3	32.0	33.6	34.8	36.1	37.2	34.5	35.9	37.4	38.8
	9/32	23.2	25.2	26.5	28.1	30.6	32.8	34.8	36.6	38.2	39.6	40.8	42.0	40.0	41.7	43.4	45.1
	5/16	26.0	28.2	29.5	31.4	34.0	36.4	38.5	40.2	41.9	43.3	44.6	45.8	44.6	46.5	48.5	50.3
	11/32	28.2	30.5	31.9	33.8	36.6	39.0	41.0	43.2	44.8	46.0	47.2	48.5	48.3	50.3	52.4	54.3
	3/8	30.9	33.5	35.3	34.5	38.1	41.2	44.0	46.7	49.2	51.5	53.5	55.5	53.6	56.0	58.2	60.5
¾	7/16				37.8	41.7	45.0	48.0	51.0	53.6	56.0	58.2	60.1				
	15/64	33.7	36.4	38.8	41.1	45.0	48.6	52.0	55.1	58.1	60.4	62.6	64.8	58.4	60.9	63.5	65.8
	1/2	38.8	42.0	44.3	47.2	51.8	55.8	59.3	62.6	65.8	68.6	70.9	73.3	67.2	70.0	73.0	75.8
	9/16	43.5	47.1	50.1	52.7	57.8	62.1	65.8	69.5	72.6	75.7	78.2	80.4	75.2	78.4	81.6	84.8
	5/8	47.6	51.4	54.6	57.5	62.4	67.2	71.3	75.0	78.4	81.4	83.6	86.3	82.1	85.5	89.2	92.5
	11/16	54.1	58.5	61.9	65.9	70.8	75.5	79.7	83.6	87.0	90.0	92.6	95.2	93.0	96.9	101.0	104.7
	1/2	58.8	63.5	67.9	67.4	74.0	80.4	85.8	90.8	95.8	100.0	104.0	107.8	101.9	106.1	110.5	114.7
1	57/64				71.8	79.0	85.2	91.1	96.6	101.8	106.0	110.2	114.1				
	7/8	66.0	71.3	76.1	80.4	88.4	95.5	101.7	107.3	112.7	117.7	122.1	126.2	114.2	119.0	124.0	128.7
	15/16	72.7	78.5	83.6	88.5	97.0	104.1	111.0	117.5	123.0	128.2	133.0	137.2	125.7	131.0	136.4	141.5
	1/2	84.5	91.4	97.0	102.4	112.0	120.0	127.2	134.3	140.5	146.0	152.0	155.1	145.9	152.0	158.2	164.1
	1/8	93.8	101.5	108.0	113.9	124.0	132.0	140.0	147.0	153.5	159.0	164.2	168.4	161.8	168.5	175.5	181.9
	1/4	103.0	111.6	118.2	114.0	124.0	132.1	143.9	155.8	164.5	171.0	179.0	185.1	179.9	187.3	195.0	202.1
1¼	7/8				126.1	139.0	151.0	161.0	171.1	180.5	188.0	196.1	203.3				
	1/8	112.7	122.0	130.8	137.5	152.8	164.1	175.0	185.5	196.0	204.0	212.4	220.6	196.3	204.5	213.0	221.3
	1/4	122.0	132.1	141.2	149.0	164.0	177.7	188.9	201.0	211.0	223.0			212.5	221.6		
	3/8	132.0	150.9	161.5	169.8	187.0	191.0	213.9	228.0								
	1/2	154.0	166.5	176.0	187.8	206.0	222.0										
	5/8	176.8	191.5	205.0	214.5												
	3/4	199.0	215.5	221.0													
1½	1/4																
	13/16	217.5															

BRANCH IS MORE THAN 8 FEET LONG, MEASURED HORIZONTALLY FROM ORIFICE ONE SIZE LARGER THAN SPECIFIED

OF BOILER IN FEET																SECTION I		
CONNECT TO RISERS ACCORDING TO FIGS. 5A-5C.																	DRILL SIZE	PIPE SIZE
17-18	18-20	20-22	22-24	24-26	26-28	28-30	30-33	33-36	36-40	40-43	43-46	46-50	50-55	55-60				
4.0	4.2	4.5	4.7	4.9	5.2	5.4	5.6	5.9	6.3	6.5	6.7	7.1	7.4	7.8			7/64	1/2
4.4	4.6	4.9	5.1	5.4	5.6	5.8	6.1	6.4	6.8	7.1	7.4	7.7	8.1	8.4			33	
4.9	5.0	5.3	5.8	6.0	6.3	6.5	6.9	7.2	7.6	7.9	8.2	8.6	9.0	9.4			32	
5.5	5.9	6.2	6.6	6.9	7.2	7.4	7.8	8.2	8.7	9.0	9.4	9.8	10.3	10.8			31	
6.0	6.4	6.8	7.1	7.5	7.8	8.1	8.5	8.9	9.5	9.8	10.2	10.7	11.2	11.7			1/8	
7.2	7.6	8.1	8.5	8.9	9.3	9.6	10.1	10.6	11.2	11.7	12.1	12.7	13.3	13.9			30	
7.9	8.4	8.9	9.3	9.8	10.2	10.6	11.2	11.7	12.4	13.0	13.3	14.0	14.7	15.3			29	
9.0	9.6	10.1	10.6	11.2	11.6	12.1	12.7	13.3	14.1	14.7	15.2	15.9	16.7	17.5			9/64	
9.9	10.5	11.1	11.7	12.3	12.8	13.3	14.0	14.7	15.6	16.1	16.7	17.5	18.4	19.2			28	
10.7	11.4	12.0	12.6	13.2	13.7	14.3	15.1	15.8	16.8	17.4	18.1	18.9	19.8	20.8			24	
11.6	12.3	13.0	13.7	14.4	15.0	15.5	16.4	17.2	18.2	18.9	19.6	20.5	21.5	22.5			5/32	
12.6	13.4	14.2	14.9	15.6	16.2	16.9	17.8	18.7	19.8	20.5	21.3	22.3	23.4	24.5			20	
13.8	14.7	15.5	16.3	17.1	17.8	18.5	19.5	20.4	21.7	22.5	23.4	24.4	25.6	26.8			19	
15.6	16.5	17.5	18.4	19.2	20.0	20.8	21.9	23.0	24.4	25.3	26.2	27.4	28.8	30.2			11/64	
17.3	18.4	19.4	20.2	21.2	22.2	23.1	24.4	25.5	27.1	28.1	29.1	30.5	32.0	33.5			15	
19.3	20.5	21.6	22.7	23.8	24.8	25.7	27.1	28.4	30.2	31.3	32.4	34.0	35.7	37.4			3/16	
21.0	22.3	23.6	24.8	25.9	27.1	28.1	29.6	31.1	32.9	34.2	35.4	37.1	39.0	40.8			9	
22.4	23.8	25.3	26.4	27.7	28.8	29.9	31.6	33.1	35.0	36.4	37.8	39.5	41.5	43.4			13/64	
24.8	26.4	27.8	29.3	30.6	31.9	33.1	34.9	36.6	38.8	40.3	41.8	43.8	46.0	48.2			4	
28.8	30.6	32.3	34.0	35.6	37.1	38.5	40.5	42.4	44.9	46.7	48.6	50.8	53.4	55.8			7/32	
32.6	34.7	36.7	38.5	40.4	42.1	43.7	46.1	48.3	51.2	53.2	55.2	57.7	60.6	63.4			9/64	
36.4	38.8	41.0	43.1	45.0	47.0	48.8	51.5	54.0	57.2	59.4	61.6	64.5	67.7	70.8			1/4	
40.2	42.7	45.0	47.4	49.5	51.7	53.6	56.6	59.3	62.8	65.4	67.7	71.0	74.9	77.8			17/64	
46.6	49.6	52.4	55.1	58.1	60.2	62.4	65.8	69.0	73.2	75.9	78.8	82.4	86.6	90.6			9/32	
52.0	55.3	58.4	61.4	65.9	67.0	70.0	73.4	77.0	81.5	84.0	87.8	91.3	96.5	101.0			5/16	
56.2	59.8	63.2	66.4	69.5	72.5	75.3	79.4	83.2	88.2	91.5	95.0	99.3	104.3	108.2			11/32	
62.5	66.5	70.2	73.8	77.4	80.6	83.6	88.2	92.5	98.1	102.0	105.7	110.5	116.2	121.5			3/8	
																	13/32	3/4
68.0	72.4	76.4	80.4	84.1	87.6	91.0	95.8	100.4	106.5	110.7	114.7	120.0	126.1	132.0			29/64	
78.3	83.2	88.0	92.5	96.9	101.0	104.9	110.6	116.0	122.8	127.3	132.2	138.4	145.3	152.2			15/32	
87.6	93.2	98.5	103.5	108.2	113.0	117.2	123.6	129.1	137.3	142.8	148.0	154.8	162.6	170.2			17/32	
95.6	101.8	107.5	113.0	118.1	123.2	128.0	135.0	141.3	150.0	155.7	161.4	169.0	177.5	185.8			9/16	
108.3	115.2	121.8	128.0	134.0	139.7	145.0	153.0	160.0	170.0	176.2	183.0	191.4	201.0	210.5			19/32	
118.7	126.3	133.2	140.1	146.5	153.0	158.8	167.4	175.3	186.0	193.0	200.5	209.6					5/8	
																	21/32	1
133.0	141.5	149.6	157.1	164.5	171.2	178.0	188.0	196.8	208.5	216.5	224.5						11/16	
146.4	155.8	164.5	173.0	181.0	188.8	196.0	206.8	217.0									23/32	
169.9	180.6	190.8	200.6	210.0	218.5	226.3											3/4	
188.2	200.1	211.3	222.3														25/32	
209.0	222.5																13/16	
																	27/32	1 1/4
																	7/8	
																	29/32	
																	15/16	
																	31/32	
																	1 1/16	
																	1 1/8	
																	1 3/16	
																	1 1/4	1 1/2
																	1 5/16	

TABLE 2. CAPACITIES OF ORIFICES AND RADIATOR BRANCHES. IF THE RADIATOR THE MAINS OR RISERS TO THE RADIATOR, USE AN

PIPE SIZE	DRILL SIZE	SECTION 2 — HEIGHT ABOVE MIDPOINT																
		CONNECT TO MAINS ACCORDING TO FIGS. 4A-4H.																
		3-3½	3½-4	4-4½	4½-5	5-6	6-7	7-8	8-9	9-10	10-11	11-12	12-13	13-14	14-15	15-16	16-17
½	7/64					2.2	2.4	2.6	2.8	3.0	3.1	3.3	3.4	2.9	3.1	3.2	3.3
	33			2.0	2.1	2.4	2.6	2.9	3.0	3.2	3.4	3.6	3.8	3.2	3.3	3.5	3.6
	32		2.0	2.3	2.4	2.7	3.2	3.3	3.5	3.7	3.9	4.1	4.2	3.6	3.7	3.9	4.4
	31	2.1	2.3	2.6	2.7	3.0	3.4	3.6	3.9	4.1	4.4	4.6	4.8	4.1	4.3	4.4	4.6
	1/8	2.3	2.5	2.8	2.9	3.3	3.7	3.9	4.2	4.5	4.7	5.0	5.2	4.4	4.6	4.8	5.0
	30	2.7	3.0	3.2	3.5	4.0	4.4	4.7	5.0	5.4	5.6	5.9	6.2	5.3	5.5	5.7	6.0
	29	2.9	3.3	3.6	3.8	4.3	4.8	5.1	5.5	5.8	6.2	6.5	6.8	5.8	6.1	6.3	6.6
	9/64	3.4	3.8	4.1	4.4	5.0	5.5	6.0	6.3	6.7	7.1	7.4	7.7	6.6	6.9	7.2	7.5
	26	3.7	4.1	4.5	4.8	5.4	6.0	6.5	6.8	7.4	7.8	8.2	8.5	7.3	7.6	7.9	8.3
	24	4.0	4.5	4.9	5.2	5.9	6.5	7.0	7.5	8.0	8.4	8.8	9.2	7.8	8.2	8.6	8.9
	5/32	4.4	4.9	5.4	5.7	6.4	7.0	7.6	8.1	8.6	9.2	9.6	10.0	8.5	8.9	9.3	9.6
	20	4.8	5.3	5.8	6.1	6.9	7.6	8.2	8.8	9.4	9.9	10.4	10.8	9.2	9.7	10.1	10.5
	19	5.2	5.8	6.4	6.8	7.6	8.3	9.0	9.7	10.3	10.8	11.3	11.8	10.1	10.6	11.1	11.5
	11/64	5.9	6.5	7.2	7.6	8.5	9.4	10.1	10.8	11.5	12.1	12.7	13.3	11.4	11.9	12.4	12.9
	15	6.5	7.2	8.0	8.4	9.5	10.4	11.2	12.0	12.8	13.4	14.1	14.7	12.7	13.4	13.8	14.4
	3/16	7.3	8.1	8.7	9.4	10.5	11.6	12.5	13.4	14.1	14.9	15.7	16.3	14.1	14.8	15.4	16.0
	9	8.0	8.8	9.6	10.2	11.5	12.6	13.6	14.5	15.4	16.2	17.0	17.6	15.4	16.1	16.8	17.5
	13/64	8.5	9.4	10.2	10.9	12.3	13.4	14.5	15.5	16.5	17.2	18.1	19.1	16.4	17.2	17.9	18.6
	4	9.4	10.4	11.4	12.0	13.5	14.9	16.0	17.1	18.1	19.0	20.0	20.7	18.2	19.0	19.9	20.6
	7/32	10.9	12.1	13.0	13.9	15.7	17.1	18.5	19.7	20.8	21.8	22.9	23.7	20.9	22.1	23.0	24.0
	15/64	12.4	13.7	14.8	15.8	17.8	19.4	20.9	22.2	23.5	24.8	25.7	26.8	23.9	25.0	26.1	27.2
	1/4	13.9	15.3	16.6	17.6	19.8	21.6	23.1	24.5	25.9	27.0	28.3	29.4	25.7	26.9	28.2	29.5
	17/64	15.3	16.9	18.3	19.5	21.8	23.7	25.3	26.9	28.4	29.5	30.8	31.7	29.4	30.7	32.2	33.4
	9/32	17.8	19.7	21.2	22.5	25.2	27.3	29.0	30.7	32.2	33.6	34.9	35.8	34.2	35.7	37.4	38.8
	5/16	20.0	22.0	23.7	25.2	28.0	30.2	32.2	33.8	35.4	36.7	38.2	39.0	38.1	40.0	41.6	43.3
	11/32	21.6	23.9	25.8	27.3	30.1	32.4	34.3	36.3	37.6	39.0	40.3	41.3	41.2	43.0	45.0	46.2
	3/8	23.8	26.2	28.4	27.6	31.3	34.4	36.8	39.2	41.4	43.5	45.6	47.4	45.9	48.0	50.1	52.1
¾	5/16				30.4	34.2	37.4	40.2	42.8	45.2	47.4	49.4	51.4				
	21/64	25.9	28.5	31.0	33.1	37.0	40.4	43.4	46.2	49.0	51.3	53.4	55.4	49.6	52.1	54.5	56.5
	11/32	29.8	32.9	35.7	38.2	42.6	46.4	49.6	52.5	55.2	58.0	60.5	62.5	57.4	60.0	62.6	65.2
	3/8	33.4	36.8	39.7	42.2	47.4	51.6	55.2	58.4	61.4	64.1	66.6	68.9	64.2	67.1	70.2	72.8
	13/32	36.6	40.4	43.3	46.0	51.2	56.0	59.7	63.0	66.0	68.8	71.3	73.7	70.2	73.4	76.5	79.6
	7/16	41.5	45.9	49.1	52.1	57.8	62.6	66.6	70.2	73.5	76.0	78.8	81.2	78.4	83.0	86.6	90.1
	½	45.2	49.8	53.9	54.1	60.8	66.3	71.7	76.3	80.6	84.5	88.6	92.2	87.0	91.0	95.0	98.5
1	7/16				57.6	64.8	70.8	76.2	81.2	85.5	89.5	93.6	97.4				
	15/32	50.7	55.9	60.2	65.0	72.5	79.2	84.9	90.0	95.4	99.5	104.1	107.8	97.6	102.0	106.4	110.8
	8/32	55.8	61.6	66.2	71.0	79.5	86.6	92.9	98.7	103.9	108.3	113.1	117.1	107.2	112.3	117.2	121.5
	½	64.9	71.7	77.3	82.3	91.7	99.6	106.5	112.8	118.4	123.5	128.0	132.4	124.6	130.4	135.9	141.2
	9/16	72.1	79.5	85.5	90.8	101.0	109.7	118.5	122.4	129.4	134.6	140.0	143.8	138.1	144.4	150.8	156.6
	5/8	79.4	87.7	95.0	91.9	103.0	113.5	123.2	131.8	138.7	144.8	151.7	158.7	153.4	160.2	167.5	174.0
1¼	17/32				101.5	114.3	125.3	135.1	143.7	152.1	159.4	167.4	173.8				
	9/16	86.8	95.8	104.7	111.0	125.2	136.3	147.1	156.5	165.3	173.5	181.5	188.6	167.9	175.0	182.9	190.0
	19/32	94.4	103.6	113.3	120.0	134.5	147.4	158.3	168.8	178.0	187.0	195.0	203.5	181.8	189.8	198.1	205.8
	5/8	107.0	118.4	128.0	136.5	153.0	167.3	179.3	191.0	201.9	210.0	219.0		206.3	216.3	255.7	
	11/16	118.6	130.9	141.1	150.8	169.0	184.5	199.2	210.0	220.0							
	¾	136.3	150.5	163.2	173.5	193.4	209.9	224.0									
	7/8	152.9	168.9	183.0	185.2	201.0	216.0										
1½	¾				195.2												
	15/16	166.9	184.7	202.7	216.5												
	7/8	189.4	209.9														
	1.	204.2															

BRANCH IS MORE THAN 8 FEET LONG, MEASURED HORIZONTALLY FROM ORIFICE ONE SIZE LARGER THAN SPECIFIED

OF BOILER IN FEET — SECTION 2

CONNECT TO RISERS ACCORDING TO FIGS. 5A-5C.

17-18	18-20	20-22	22-24	24-26	26-28	28-30	30-33	33-36	36-40	40-43	43-46	46-50	50-55	55-60	DRILL SIZE	PIPE SIZE
3.4	3.7	3.9	4.1	4.3	4.5	4.7	4.9	5.2	5.5	5.7	5.9	6.2	6.5	6.8	7/64	½
3.8	4.0	4.3	4.5	4.7	4.9	5.1	5.4	5.6	6.0	6.2	6.4	6.7	7.1	7.4	33	
4.2	4.5	4.8	5.0	5.2	5.5	5.7	6.0	6.4	6.8	7.0	7.2	7.5	7.9	8.3	32	
4.8	5.1	5.4	5.7	6.0	6.2	6.5	6.8	7.2	7.6	7.9	8.2	8.6	9.0	9.5	31	
5.2	5.6	5.9	6.2	6.5	6.8	7.1	7.4	7.8	8.3	8.6	8.9	9.3	9.8	10.3	1/8	
6.2	6.6	7.0	7.4	7.7	8.1	8.4	8.9	9.3	9.8	10.2	10.6	11.1	11.7	12.2	30	
6.8	7.3	7.7	8.1	8.5	8.9	9.2	9.8	10.2	10.8	11.3	11.7	12.2	12.9	13.5	29	
7.8	8.3	8.8	9.2	9.7	10.1	10.5	11.1	11.6	12.3	12.8	13.3	13.9	14.7	15.3	9/64	
8.5	9.1	9.6	10.2	10.7	11.1	11.6	12.2	12.8	13.6	14.1	14.6	15.3	16.1	16.9	26	
9.2	9.8	10.4	11.0	11.5	12.0	12.5	13.2	13.8	14.7	15.2	15.8	16.5	17.4	18.2	24	
10.0	10.7	11.3	11.9	12.5	13.1	13.6	14.3	15.0	15.9	16.5	17.1	17.9	18.9	19.8	5/32	
10.8	11.6	12.3	12.9	13.6	14.2	14.7	15.5	16.3	17.3	17.9	18.7	19.5	20.5	21.4	20	
11.9	12.7	13.4	14.2	14.9	15.5	16.1	17.0	17.9	18.9	19.7	20.4	21.4	22.5	23.5	19	
13.4	14.3	15.1	16.0	16.8	17.5	18.2	19.2	20.1	21.3	22.2	23.0	24.0	25.3	26.5	11/64	
14.9	15.9	16.8	17.7	18.6	19.2	20.0	21.1	22.1	23.6	24.6	25.5	26.7	28.1	29.4	15	
16.6	17.7	18.7	19.8	20.8	21.6	22.5	23.9	24.9	26.4	27.4	28.4	29.8	31.3	32.8	3/16	
18.1	19.3	20.4	21.5	22.5	23.4	24.5	25.9	26.9	28.8	29.9	31.0	32.4	34.2	35.8	9	
19.3	20.6	21.7	22.8	24.1	25.1	26.2	27.6	28.9	30.6	31.8	32.0	34.6	36.4	38.1	13/64	
21.4	22.8	24.1	25.5	26.7	27.9	29.0	30.6	32.0	34.0	35.3	36.6	38.3	40.4	42.2	4	
24.8	26.5	28.0	29.5	31.1	32.3	33.6	35.5	37.2	39.4	41.0	42.5	44.5	46.9	49.1	7/32	
28.2	30.0	31.8	33.5	35.2	36.7	38.2	40.2	42.2	44.8	46.5	48.2	50.5	53.2	55.6	9/64	
30.7	33.0	35.5	37.5	39.3	41.0	42.6	45.0	47.1	50.1	52.0	53.9	56.4	59.4	62.2	1/4	
34.6	36.9	39.0	41.2	43.3	45.2	46.9	49.5	51.9	55.1	57.2	59.3	62.1	65.4	68.5	17/64	
40.2	42.7	45.3	47.8	50.3	52.4	54.5	57.5	60.3	64.0	66.5	68.9	72.2	76.0	79.5	9/32	
44.8	48.5	50.5	53.4	56.1	58.4	61.3	65.0	68.0	71.3	74.2	76.9	80.5	84.7	88.7	5/16	
48.5	51.8	54.7	57.7	60.8	63.2	65.7	69.4	72.6	77.1	80.1	83.1	87.0	91.6	95.8	11/32	
53.8	57.5	60.8	64.4	67.4	70.3	73.1	77.1	80.8	85.7	89.1	92.4	96.7	101.7	106.5	3/8	
															25/64	¾
58.6	62.6	66.2	69.8	73.4	76.5	79.5	83.9	88.0	93.2	96.5	100.3	105.0	110.4	115.7	13/32	
67.4	72.0	76.3	80.4	84.4	88.0	91.5	96.5	101.2	107.1	111.3	115.5	120.8	127.3	133.4	17/32	
75.5	80.6	85.3	90.0	94.5	98.5	102.3	108.0	113.2	120.0	124.7	129.2	134.6	141.5	146.5	9/16	
82.4	87.9	93.1	98.3	103.0	107.5	111.9	118.0	123.5	131.0	136.0	141.0	147.8	155.5	163.0	19/32	
93.3	99.8	105.3	114.8	116.8	121.8	126.8	133.7	140.0	148.4	154.1	159.9	167.3	176.1	184.5	5/8	
102.1	109.2	115.5	121.8	128.0	133.5	138.8	146.2	153.3	162.3	168.0	174.8	183.1	192.9	202.0	½	
															57/64	1
114.6	122.4	129.5	136.6	143.5	149.6	155.5	164.0	172.0	182.0	189.2	196.3	205.7	216.1		29/32	
126.2	134.8	142.7	150.5	157.8	164.6	171.0	180.5	189.0	200.2	208.2	216.0				15/16	
146.2	156.2	165.3	174.0	183.0	191.0	198.4	209.3	219.4							1/2	
162.3	173.1	183.1	193.5	203.0	211.5	220.0									5/8	
180.1	192.1	203.8	214.3												3/8	
															17/32	1¼
197.0	210.2														9/16	
213.0															19/32	
															5/8	
															11/16	
															¾	
															7/8	
															¾	1½
															13/16	
															7/8	
															1.	

TABLE 2. CAPACITIES OF ORIFICES AND RADIATOR BRANCHES. IF THE RADIATOR THE MAINS OR RISERS TO THE RADIATOR, USE AN

PIPE SIZE	DRILL SIZE	SECTION. 3									HEIGHT ABOVE MIDPOINT							
		CONNECT TO MAINS ACCORDING TO FIGS. 4A-4N.																
		3-3½	3½-4	4-4½	4½-5	5-6	6-7	7-8	8-9	9-10	10-11	11-12	12-13	13-14	14-15	15-16	16-17	
½	7/64							2.0	2.1	2.3	2.4	2.5	2.7	2.3	2.4	2.5	2.6	
	33						2.0	2.1	2.3	2.5	2.8	2.7	2.9	2.5	2.6	2.7	2.8	
	32					2.0	2.2	2.4	2.6	2.8	3.0	3.1	3.2	2.8	2.9	3.0	3.2	
	31				2.0	2.2	2.5	2.7	3.0	3.1	3.3	3.5	3.7	3.2	3.3	3.5	3.6	
	1/8				2.1	2.4	2.8	3.0	3.2	3.4	3.6	3.8	4.0	3.4	3.5	3.8	3.9	
	30		2.0	2.3	2.5	2.9	3.2	3.5	3.8	4.0	4.3	4.5	4.8	4.1	4.3	4.5	4.7	
	29	2.0	2.2	2.5	2.8	3.2	3.6	3.9	4.2	4.5	4.8	4.9	5.3	4.5	4.7	4.9	5.2	
	9/64	2.2	2.6	2.9	3.1	3.6	4.1	4.4	4.8	5.1	5.3	5.6	6.0	5.1	5.4	5.7	5.9	
	26	2.4	2.8	3.1	3.4	4.0	4.4	4.9	5.3	5.6	5.9	6.1	6.6	5.6	5.9	6.2	6.5	
	24	2.6	3.0	3.4	3.7	4.3	4.8	5.2	5.7	6.0	6.4	6.7	7.1	6.1	6.4	6.7	7.0	
	5/32	2.9	3.3	3.7	4.0	4.7	5.3	5.8	6.2	6.6	7.0	7.4	7.7	6.8	6.8	7.4	7.6	
	20	3.1	3.6	4.0	4.4	5.1	5.6	6.2	6.7	7.1	7.5	8.0	8.3	7.2	7.5	7.9	8.2	
	19	3.4	3.9	4.4	4.8	5.6	6.2	6.8	7.3	7.8	8.3	8.8	9.1	7.9	8.2	8.6	9.0	
	11/64	3.8	4.4	5.0	5.4	6.3	7.0	7.6	8.1	8.7	9.3	9.7	10.2	8.9	9.3	9.7	10.1	
	15	4.2	4.9	5.5	6.0	6.9	7.7	8.4	9.1	9.7	10.2	10.8	11.3	9.8	10.3	10.7	11.3	
	3/16	4.7	5.5	6.1	6.7	7.7	8.6	9.4	10.2	10.8	11.4	11.9	12.6	10.9	11.5	11.9	12.6	
	9	5.2	5.9	6.8	7.3	8.4	9.5	10.3	11.0	11.6	12.3	12.9	13.6	11.9	12.5	13.1	13.6	
	13/64	5.5	6.4	7.1	7.8	9.0	10.0	10.9	11.7	12.4	13.2	14.0	14.6	12.7	13.3	13.9	14.6	
	4	6.1	7.1	7.9	8.6	9.9	11.0	12.1	13.0	13.7	14.5	15.2	15.9	14.1	14.8	15.5	16.2	
	7/32	7.1	8.1	9.1	10.0	11.5	12.7	13.9	15.0	15.8	16.7	17.5	18.3	16.4	17.2	18.0	18.8	
	15/64	8.1	9.3	10.4	11.4	13.2	14.4	15.7	16.8	17.8	18.8	19.6	20.5	18.6	19.5	20.4	21.3	
	1/4	9.0	10.4	11.6	12.7	14.5	16.0	17.4	18.5	19.6	20.6	21.6	22.6	20.8	21.8	22.8	23.8	
	17/64	10.0	11.5	12.8	14.0	15.9	17.6	19.0	20.4	21.5	22.6	23.6	24.5	22.8	23.9	25.0	26.2	
	9/32	11.6	13.4	14.8	16.2	18.4	20.2	21.8	23.2	24.4	25.6	26.6	27.7	26.5	27.8	29.1	30.4	
	5/16	13.0	15.0	16.6	18.0	20.3	22.4	24.2	25.7	27.0	28.1	29.0	30.4	29.6	31.0	32.5	33.9	
	11/32	14.0	16.3	18.0	19.7	22.4	24.5	26.2	27.6	28.8	29.8	30.4	31.9	32.0	33.6	35.1	36.6	
	3/8	15.5	17.8	19.9	19.8	22.9	25.3	27.5	29.8	31.4	33.4	34.4	36.5	35.6	37.3	39.0	40.8	
¾	5/16				21.7	25.1	27.7	30.2	32.3	34.1	35.9	37.9	39.4					
	9/64	16.8	19.4	21.6	23.6	27.0	30.0	32.6	35.2	37.1	39.1	41.0	42.7	38.7	40.6	42.5	44.3	
	11/32	19.4	22.4	25.0	27.1	31.2	34.4	37.2	40.0	41.9	44.5	46.2	48.2	44.5	46.7	48.9	51.0	
	3/8	21.7	25.1	27.8	30.2	34.7	38.2	41.4	44.0	46.3	48.8	51.0	53.1	49.8	52.3	54.7	57.0	
	13/32	23.7	27.4	30.4	32.9	37.5	41.6	44.8	47.8	50.3	52.7	54.8	56.8	53.6	57.1	59.7	62.3	
	7/16	26.9	31.2	34.4	37.2	42.5	46.5	50.0	53.3	56.0	58.4	60.5	62.8	61.6	64.6	67.6	70.6	
	½	29.3	33.9	37.8	38.8	44.6	49.2	53.9	58.0	61.8	64.9	68.2	71.1	67.5	70.8	74.0	77.5	
1	9/16				41.3	47.5	52.6	57.2	61.0	65.0	68.6	71.8	75.2					
	5/8	32.9	38.0	42.4	46.0	53.1	58.7	63.8	68.5	72.0	76.4	79.5	83.1	75.8	79.5	83.0	86.7	
	11/16	36.2	41.9	46.5	50.9	58.3	64.2	69.6	74.8	78.7	83.2	86.8	90.3	83.4	87.4	91.5	95.5	
	½	42.1	48.1	54.0	59.0	67.2	73.9	80.0	85.0	89.4	94.6	97.5	102.1	96.5	101.2	106.0	110.9	
	9/16	46.8	54.1	60.0	65.5	74.0	81.4	87.9	93.2	98.2	103.0	107.0	110.9	106.0	111.4	117.5	121.8	
	3/8	51.5	59.6	66.5	69.8	76.0	85.2	91.7	99.4	104.8	111.0	116.8	122.9	113.0	125.0	130.5	136.0	
1¼	7/16				72.7	83.6	92.8	101.3	108.5	115.2	122.0	127.0	132.3					
	9/16	56.3	65.2	72.8	79.3	91.8	101.2	110.2	118.0	125.2	132.5	138.5	144.5	130.1	136.2	143.0	149.0	
	11/16	61.4	70.5	77.6	84.6	97.5	108.2	118.9	127.5	134.6	142.5	149.0	156.2	140.8	147.1	154.8	159.6	
	3/8	69.5	80.6	89.8	97.6	112.3	124.0	134.6	143.5	152.0	161.0	167.5	175.3	160.8	168.1	176.1	183.6	
	1/16	77.5	89.6	98.2	108.2	124.0	136.8	149.8	159.0	168.2	175.7	183.5	191.0	177.8	186.5	195.0	203.2	
	1/4	88.5	102.5	114.2	123.8	141.9	155.8	168.1	178.5	188.6	197.0	206.0	214.5	203.9	214.0	222.4		
	3/16	98.3	114.9	128.0	138.8	144.7	160.0	174.0	186.0	198.0	208.3	219.3						
1½	5/16				140.0	160.7	179.0	195.0	209.0	221.0								
	11/16	108.3	125.7	139.9	152.2	174.8	192.6	209.0										
	7/16	123.5	143.4	158.2	173.0	197.0	216.0											
	1	132.7	154.2	171.4	186.7	213.2												
	1 1/16	145.3	168.1	186.0	202.0													
	1 1/8	162.2	187.3	206.0														
**	1 1/4	184.7	214.0															
	1 3/8	212.2																

** RADIATOR BRANCHES CONSIST OF 1½" RISERS WITH 2" CONNECTIONS TO

BRANCH IS MORE THAN 8 FEET LONG, MEASURED HORIZONTALLY FROM ORIFICE ONE SIZE LARGER THAN SPECIFIED

OF BOILER IN FEET — SECTION 3

CONNECT TO RISERS ACCORDING TO FIGS. 5A - 5C.

17-18	18-20	20-22	22-24	24-26	26-28	28-30	30-33	33-36	36-40	40-43	43-46	46-50	50-55	55-60	DRILL SIZE	PIPE SIZE
2.7	2.9	3.1	3.2	3.4	3.5	3.7	3.9	4.1	4.4	4.5	4.7	4.9	5.2	5.4	7/64	1/2
2.9	3.2	3.3	3.5	3.7	3.9	4.0	4.3	4.5	4.7	4.9	5.1	5.4	5.7	5.9	33	
3.3	3.5	3.8	3.9	4.1	4.3	4.5	4.8	5.0	5.3	5.5	5.7	6.0	6.3	6.6	32	
3.7	4.0	4.3	4.5	4.7	4.9	5.1	5.4	5.7	6.0	6.3	6.5	6.9	7.2	7.6	31	
4.1	4.4	4.6	4.9	5.1	5.4	5.6	5.9	6.2	6.6	6.9	7.1	7.5	7.8	8.2	1/8	
4.9	5.2	5.5	5.8	6.1	6.4	6.6	7.0	7.4	7.8	8.1	8.5	8.9	9.3	9.8	30	
5.3	5.7	6.1	6.4	6.7	7.0	7.3	7.7	8.1	8.6	9.0	9.3	9.8	10.3	10.8	29	
6.1	6.5	6.9	7.3	7.7	8.0	8.3	8.8	9.3	9.8	10.2	10.6	11.1	11.7	12.3	9/64	
6.7	7.2	7.6	8.0	8.4	8.8	9.2	9.7	10.2	10.8	11.3	11.7	12.2	12.9	13.5	26	
7.2	7.7	8.2	8.7	9.1	9.5	9.9	10.5	11.0	11.7	12.2	12.6	13.2	13.9	14.6	24	
7.8	8.4	8.9	9.3	9.9	10.3	10.7	11.4	11.9	12.7	13.2	13.7	14.3	15.1	15.8	5/32	
8.5	9.2	9.7	10.2	10.7	11.2	11.7	12.3	13.0	13.7	14.3	14.9	15.6	16.4	17.2	20	
9.3	10.0	10.6	11.2	11.8	12.3	12.8	13.5	14.2	15.1	15.7	16.3	17.1	17.9	18.8	19	
10.5	11.3	12.0	12.6	13.2	13.8	14.4	15.2	16.0	17.0	17.7	18.3	19.2	20.2	21.2	1/64	
11.7	12.5	13.3	14.0	14.7	15.3	16.0	16.9	17.8	18.8	19.6	20.3	21.3	22.4	23.5	18	
13.0	14.0	14.8	15.6	16.4	17.1	17.8	18.8	19.8	21.0	21.9	22.7	23.8	25.0	26.3	3/16	
14.2	15.2	16.2	17.1	18.0	18.6	19.4	20.5	21.4	22.7	23.7	24.8	25.9	27.3	28.6	9	
15.2	16.2	17.2	18.2	19.0	19.9	20.7	21.9	23.0	24.4	25.4	26.4	27.6	29.1	30.3	1/64	
16.8	18.0	19.1	20.2	21.3	22.0	22.9	24.2	25.5	27.0	28.2	29.2	30.6	32.2	33.8	4	
19.4	20.8	22.1	23.3	24.5	25.6	26.8	28.1	29.6	31.4	32.7	33.9	35.5	37.4	39.2	7/32	
22.1	23.6	25.1	26.5	27.8	29.0	30.2	31.9	33.6	35.6	37.1	38.5	40.3	42.5	44.5	9/64	
24.7	26.5	28.1	29.6	31.0	32.4	33.8	35.7	37.5	39.8	41.5	43.0	45.0	47.4	49.6	5	
27.1	29.1	30.8	32.5	34.1	35.7	37.1	39.3	41.3	43.8	45.7	47.3	49.6	52.2	54.6	11/64	
31.6	33.8	35.9	37.9	39.6	41.5	43.2	45.6	48.0	50.9	53.0	55.0	57.6	60.7	63.7	9/32	
35.1	37.7	40.1	42.2	44.2	46.3	48.0	50.9	53.5	56.7	59.2	61.3	64.3	67.7	71.0	5/16	
38.1	40.7	43.4	45.6	47.8	50.0	52.0	55.0	57.8	61.3	63.9	66.3	69.5	73.1	76.7	11/64	
42.3	45.9	48.2	50.8	53.1	55.5	57.9	61.2	64.3	68.2	71.0	73.7	77.3	81.4	85.5	3/8	
															13/32	3/4
46.0	49.3	52.4	55.1	57.8	60.5	62.9	66.5	70.0	74.2	77.3	80.2	84.0	88.5	92.9	7/64	
52.9	56.8	60.2	63.5	66.5	69.6	72.4	76.6	80.5	85.6	89.0	92.3	96.7	100.6	106.8	1/2	
59.2	63.5	67.5	71.0	74.5	77.8	81.0	85.7	88.5	95.5	98.0	101.5	106.5	113.4	119.5	3/8	
66.6	69.4	73.6	77.5	81.3	85.0	88.6	93.8	98.4	104.3	108.7	112.8	118.1	124.3	130.5	15/64	
73.3	78.4	83.4	87.8	92.0	95.4	100.3	106.0	111.5	118.1	123.1	127.8	133.7	140.8	147.5	7/16	
80.2	86.0	91.4	96.2	101.0	105.4	109.8	116.1	122.0	129.4	135.0	140.0	146.5	154.0	161.6	1/2	
															17/32	1
90.0	96.5	102.6	107.9	113.2	118.2	123.0	130.0	136.9	145.0	151.2	156.9	164.2	173.9	181.5	9/16	
99.0	106.2	113.0	118.8	125.4	130.1	135.5	143.1	150.6	159.6	166.3	172.6	180.8	190.0	199.5	19/32	
114.9	123.0	131.0	137.9	144.2	151.0	157.0	166.0	174.6	185.2	193.0	200.0	210.0	220.8	231.9	1/2	
127.1	136.4	145.0	152.5	160.0	167.1	174.1	182.8	193.4	205.3	214.0	222.0				9/16	
141.3	151.2	161.1	169.5	177.9	186.0	193.6	204.6	215.0							5/8	
															11/16	1 1/4
154.8	165.8	176.0	185.5	194.2	203.2	210.0	223.8								3/4	
167.5	179.5	190.5	201.0	210.3	219.2										13/16	
191.0	204.0	217.0													7/8	
211.0															15/16	
															1	
															1 1/16	
															1 1/8	
															1 3/16	**
															1 1/4	

THE MAINS. THE REDUCING ELBOWS LOOK UP

TABLE 2. CAPACITIES OF ORIFICES AND RADIATOR BRANCHES. IF THE RADIATOR THE MAINS OR RISES TO THE RADIATOR, USE AN

PIPE SIZE	DRILL SIZE	SECTION 4 — HEIGHT ABOVE MIDPOINT																
		CONNECT TO MAINS ACCORDING TO FIGS. 4A - 4H.																
		3-3½	3½-4	4-4½	4½-5	5-6	6-7	7-8	8-9	9-10	10-11	11-12	12-13	13-14	14-15	15-16	16-17
½	7/64																
	33																
	32											2.0	2.1			2.0	2.1
	31									2.0	2.2	2.3	2.4	2.1	2.2	2.3	2.4
	1/8								2.1	2.2	2.4	2.5	2.6	2.3	2.4	2.5	2.6
	30							2.0	2.3	2.5	2.6	2.8	3.0	3.1	2.7	2.8	3.0	3.1
	29						2.0	2.3	2.5	2.7	2.9	3.1	3.3	3.4	3.0	3.1	3.3	3.4
	9/64						2.3	2.6	2.8	3.1	3.3	3.5	3.7	3.9	3.4	3.6	3.7	3.9
	26					2.1	2.5	2.8	3.1	3.4	3.6	3.9	4.1	4.2	3.7	3.9	4.1	4.3
	24				2.0	2.3	2.7	3.0	3.4	3.7	3.9	4.2	4.4	4.7	4.0	4.2	4.4	4.6
	5/32				2.2	2.4	3.0	3.3	3.6	4.0	4.2	4.5	4.8	5.0	4.4	4.6	4.8	5.0
	20			2.0	2.3	2.6	3.1	3.6	4.0	4.3	4.6	4.9	5.3	5.4	4.7	5.0	5.2	5.4
	19			2.2	2.6	2.9	3.5	3.9	4.3	4.7	5.1	5.3	5.7	5.9	5.2	5.5	5.7	6.0
	11/64	2.0	2.5	2.9	3.3	3.9	4.4	4.9	5.3	5.7	6.0	6.4	6.7	5.8	6.2	6.4	6.7
	15	2.2	2.8	3.2	3.6	4.3	4.9	5.4	5.8	6.3	6.7	7.0	7.4	6.5	6.8	7.1	7.5
	3/16	2.5	3.1	3.6	4.0	4.8	5.4	6.0	6.5	7.0	7.4	7.8	8.2	7.2	7.6	8.0	8.3
	9	2.7	3.4	3.9	4.4	5.2	5.9	6.5	7.1	7.6	8.0	8.4	8.9	7.9	8.3	8.7	9.1
	13/64	2.9	3.6	4.2	4.7	5.6	6.3	7.0	7.7	8.0	8.5	9.0	9.5	8.4	8.9	9.3	9.7
	4	3.2	4.0	4.6	5.2	6.2	7.0	7.7	8.4	8.9	9.4	9.9	10.4	9.3	9.8	10.3	10.7
	7/32	3.8	4.7	5.4	6.0	7.1	8.0	8.9	9.6	10.3	10.9	11.4	11.9	10.8	11.4	11.9	12.4
	5/64	4.3	5.3	6.1	6.9	8.1	9.1	10.0	10.8	11.6	12.4	12.8	13.4	12.3	12.9	13.5	14.1
	1/4	4.8	5.9	6.8	7.7	9.0	10.1	11.1	12.0	12.8	13.5	14.1	14.7	13.7	14.4	15.1	15.7
	17/64	5.3	6.5	7.5	8.4	9.9	11.1	12.2	13.1	13.9	14.6	15.1	15.5	15.1	15.9	16.6	17.4
	9/32	6.1	7.6	8.7	9.8	11.4	12.8	13.9	15.0	15.9	16.7	17.3	18.0	17.5	18.5	19.3	20.2
	5/16	6.9	8.5	9.7	10.9	12.7	14.2	15.4	16.5	17.4	18.2	19.1	19.6	19.5	20.6	21.5	22.5
	11/32	7.4	9.2	10.5	11.7	13.7	15.2	16.5	17.7	18.5	19.3	20.1	20.8	21.1	22.2	23.3	24.3
	3/8	8.2	10.1	11.7	12.0	14.2	16.1	17.7	19.2	20.4	21.6	22.8	23.8	23.4	24.7	25.8	27.0
¾	5/16				13.1	15.5	17.6	19.3	20.8	22.2	23.5	24.8	25.9				
	21/64	8.9	11.0	12.7	14.2	16.8	19.0	20.9	22.5	24.1	25.4	26.7	27.8	25.5	26.9	28.2	29.4
	11/32	10.3	12.6	14.6	16.4	19.3	21.8	23.8	25.9	27.4	28.9	30.2	31.4	29.4	31.0	32.4	33.8
	3/8	11.5	14.2	16.3	18.2	21.5	24.2	26.4	28.4	30.1	31.7	33.3	34.8	32.9	34.6	36.2	37.8
	13/32	12.6	15.6	17.8	19.8	23.2	26.2	28.6	30.7	32.5	34.2	35.6	37.1	35.9	37.2	39.6	41.3
	7/16	14.3	17.7	20.2	22.4	26.4	29.5	32.0	34.2	36.2	37.8	39.5	40.8	40.7	42.9	44.6	46.6
	1/2	15.5	19.3	22.3	23.4	27.6	31.1	34.4	37.2	39.8	42.1	44.3	46.4	44.5	47.0	49.1	51.2
1	7/16				24.9	29.4	33.2	36.6	39.6	42.2	44.7	46.9	49.0				
	15/32	17.4	21.6	24.8	27.9	32.9	37.2	40.7	43.9	46.9	49.5	52.0	54.3	49.9	52.6	55.0	57.4
	31/64	19.2	23.8	27.2	30.7	36.2	40.7	44.7	48.2	51.2	54.0	56.5	58.9	55.0	57.9	60.6	63.2
	1/2	22.3	27.4	31.6	35.6	41.7	46.7	51.1	55.0	58.3	61.5	64.5	66.7	63.8	67.2	70.4	73.3
	9/16	24.8	30.7	35.2	39.4	46.2	51.5	56.0	59.9	63.6	66.8	69.8	72.4	70.7	74.5	78.0	81.3
	5/8	27.4	33.9	39.0	43.8	47.0	53.3	58.6	63.8	68.2	72.3	76.2	80.2	78.6	82.8	86.8	90.4
1¼	7/16				44.0	51.9	58.8	64.9	70.1	74.9	79.1	83.5	87.6				
	9/16	29.9	37.0	42.8	48.0	56.9	64.0	70.5	76.4	81.4	86.0	90.4	95.0	85.9	90.5	94.7	98.7
	19/32	32.6	39.6	46.1	52.0	61.2	69.2	76.2	82.4	87.7	93.0	97.2	102.1	93.0	98.0	102.5	107.0
	5/8	37.0	45.6	52.7	59.2	69.5	78.6	86.3	93.3	99.0	104.3	109.2	114.7	106.0	111.6	116.8	121.7
	11/16	41.0	50.7	58.1	65.5	76.8	86.6	95.0	102.0	108.0	113.8	119.3	125.0	117.2	123.5	129.0	134.6
	3/4	47.1	58.5	67.0	74.9	87.7	98.2	108.3	115.6	122.1	128.5	134.7	140.7	134.6	142.0	148.2	154.8
	7/8	52.6	64.2	75.0	75.9	96.6	101.3	111.3	120.5	127.8	136.0	142.9	149.6	151.5	159.0	166.0	173.5
1½	3/4				84.3	99.5	112.4	123.0	132.8	141.5	149.6	157.7	164.3				
	13/16	57.4	71.3	81.8	92.0	108.6	122.1	133.8	144.6	154.0	162.1	170.0	177.5	164.1	173.0	181.0	189.0
	7/8	65.2	80.0	92.6	103.2	122.4	137.4	149.3	162.0	172.0	180.9	189.5	197.0	186.0	196.0	205.0	214.0
	1	70.7	86.7	100.3	112.8	132.5	148.5	161.5	173.0	184.0	193.3	202.5	210.0	201.5	212.0	222.0	
	1⅛	76.8	93.4	108.2	122.2	143.4	160.2	174.5	187.0	197.5	206.3	214.8		218.5			
	1⅜	85.9	105.0	121.3	136.2	158.0	177.0	192.0	204.2	216.0							
**	1⅛	97.7	121.5	138.0	152.0	176.0	196.0	213.5									
	1⅜	117.3	145.9	165.6	183.0	209.0	225.0										

** RADIATOR BRANCHES CONSIST OF 1½" RISERS WITH 2" CONNECTIONS TO THE MAINS.

BRANCH IS MORE THAN 8 FEET LONG, MEASURED HORIZONTALLY FROM ORIFICE ONE SIZE LARGER THAN SPECIFIED

OF BOILER IN FEET — SECTION 4

CONNECT TO RISERS ACCORDING TO FIGS. 5A - 5C.

17-18	18-20	20-22	22-24	24-26	26-28	28-30	30-33	33-36	36-40	40-43	43-46	46-50	50-55	55-60	DRILL SIZE	PIPE SIZE
		2.0	2.2	2.3	2.4	2.5	2.7	2.8	2.9	3.0	3.2	3.3	3.5	3.7	7/64	1/2
2.0	2.1	2.2	2.4	2.5	2.6	2.7	2.9	3.0	3.2	3.3	3.4	3.6	3.8	4.0	33	
2.2	2.3	2.5	2.6	2.8	2.9	3.0	3.2	3.4	3.6	3.7	3.9	4.0	4.2	4.5	32	
2.5	2.7	2.8	3.0	3.2	3.3	3.4	3.6	3.8	4.1	4.2	4.4	4.6	4.8	5.1	31	
2.7	2.9	3.1	3.3	3.4	3.7	3.8	4.0	4.2	4.4	4.6	4.8	5.0	5.2	5.5	1/8	
3.2	3.5	3.7	3.9	4.1	4.3	4.5	4.7	5.0	5.3	5.5	5.7	6.0	6.1	6.6	30	
3.5	3.8	4.1	4.3	4.5	4.7	4.9	5.2	5.5	5.8	6.0	6.3	6.6	6.9	7.2	29	
4.0	4.3	4.6	4.9	5.1	5.4	5.6	5.9	6.2	6.6	6.9	7.1	7.5	7.8	8.3	9/64	
4.4	4.8	5.1	5.4	5.6	5.9	6.1	6.5	6.8	7.3	7.6	7.9	8.2	8.6	9.1	26	
4.8	5.2	5.5	5.8	6.1	6.4	6.6	7.0	7.4	7.8	8.2	8.5	8.9	9.3	9.8	24	
5.2	5.6	6.0	6.3	6.6	6.9	7.2	7.6	8.0	8.5	8.9	9.2	9.6	10.1	10.6	3/32	
5.7	6.1	6.5	6.8	7.2	7.5	7.8	8.3	8.7	9.2	9.6	10.0	10.5	10.9	11.5	20	
6.2	6.7	7.1	7.5	7.9	8.2	8.6	9.1	9.5	10.1	10.5	10.9	11.5	12.0	12.7	19	
7.0	7.5	8.0	8.4	8.8	9.3	9.7	10.2	10.7	11.4	11.9	12.3	12.9	13.5	14.3	11/64	
7.8	8.3	8.9	9.3	9.7	10.2	10.7	11.4	11.9	12.7	13.2	13.7	14.3	14.9	15.7	15	
8.6	9.3	9.9	10.4	11.0	11.5	12.0	12.6	13.3	14.1	14.7	15.3	16.0	16.7	17.6	3/16	
9.4	10.1	10.8	11.4	12.0	12.5	13.0	13.8	14.5	15.4	16.0	16.6	17.4	18.2	19.0	9	
10.0	10.8	11.4	12.1	12.7	13.3	13.9	14.7	15.4	16.4	17.1	17.7	18.6	19.4	20.5	13/64	
11.1	11.9	12.7	13.4	14.1	14.8	15.4	16.2	17.1	18.2	18.9	19.6	20.6	21.5	22.7	4	
12.9	13.9	14.7	15.6	16.4	17.2	17.9	18.9	19.9	21.1	22.0	22.8	23.9	25.0	26.0	7/32	
14.7	15.7	16.7	17.7	18.6	19.5	20.2	21.4	22.5	23.9	24.9	25.9	27.1	28.4	29.9	9/64	
16.4	17.6	18.7	19.8	20.8	21.8	22.6	23.9	25.2	26.7	27.8	28.9	30.3	31.6	33.4	1/4	
18.0	19.3	20.5	21.8	22.8	23.9	24.9	26.3	27.7	29.4	30.6	31.8	33.3	35.1	36.8	17/64	
20.9	22.4	23.9	25.3	26.6	27.8	29.0	30.6	32.2	34.2	35.6	36.9	38.8	40.8	42.7	9/32	
23.3	25.1	26.6	28.2	29.6	31.0	32.3	34.1	35.9	38.1	39.5	41.1	43.2	45.5	47.7	5/16	
25.2	27.1	28.8	30.5	32.0	33.5	34.9	36.8	38.8	41.2	42.8	44.5	46.6	49.2	51.5	21/64	
28.0	30.1	32.0	33.9	35.6	37.2	38.8	41.0	43.2	45.8	47.7	49.5	51.9	54.6	57.3	3/8	
															25/64	3/4
30.5	32.8	34.8	36.8	38.7	40.5	42.2	44.6	46.9	49.8	51.8	53.8	56.5	59.3	62.3	13/32	
35.1	37.6	40.1	42.4	44.5	46.6	48.5	51.3	54.0	57.3	59.7	62.0	64.9	68.3	71.7	15/32	
39.3	42.2	44.8	47.4	49.8	52.1	54.4	57.4	60.5	64.2	66.8	69.5	72.7	76.5	80.2	3/8	
42.8	46.1	49.0	51.8	54.4	57.0	59.4	62.8	66.0	70.0	73.0	75.8	79.4	83.0	87.6	13/32	
48.6	52.2	55.5	58.7	61.7	64.5	67.3	71.0	74.8	78.4	82.6	85.9	90.0	94.1	99.3	7/16	
53.3	57.2	60.8	64.3	67.5	70.8	73.7	77.8	81.9	86.9	90.5	94.1	98.4	103.0	108.5	1/2	
															57/64	1
59.7	64.0	68.2	72.1	75.7	79.2	82.5	87.2	91.8	97.5	101.3	105.3	110.3	115.3	121.8	7/8	
65.7	70.5	75.0	79.3	83.3	87.3	91.0	96.0	101.0	107.2	111.5	116.0	121.4	127.0	134.0	15/16	
76.2	81.8	86.9	92.0	96.5	101.1	105.3	111.3	117.2	124.3	129.5	134.5	141.0	147.5	155.7	1/2	
84.4	90.7	96.3	101.0	105.5	111.0	115.8	123.3	129.0	136.8	143.5	148.0	155.0	162.0	171.0	9/16	
93.8	100.8	107.0	113.3	119.0	124.7	130.0	137.1	144.5	153.2	159.0	166.0	173.9	182.0	192.0	5/8	
															19/32	1 1/4
102.5	110.0	117.1	123.9	130.1	136.2	142.0	150.0	157.7	167.5	174.5	181.1	188.9	196.5	209.5	11/16	
111.0	119.3	126.8	134.1	140.9	147.5	153.8	162.2	171.0	181.2	189.0	196.5	205.3	215.0	227.0	23/32	
126.5	136.0	144.3	152.2	160.5	168.1	175.0	185.0	194.6	206.5	215.2	223.5				3/4	
140.0	150.4	159.6	168.3	177.5	186.0	194.0	204.5	215.2							13/16	
161.0	172.6	183.4	194.1	202.0	213.5	222.1									7/8	
180.0	193.5	205.0	217.5												15/16	
															1	1 1/2
196.0	210.6	224.0													1 1/16	
222.1															1 1/8	
															1 3/16	
															1 1/4	
															1 5/16	**
															1 3/8	

THE REDUCING ELBOWS LOOK UP

number is 58.8 and by following the same line across the table to the left margin, it will be found that these "radiators" each require 1-in. "radiator" branches and a 5/8-in. orifice. In Fig. 10, the size of the drill required to bore an orifice is indicated by a number within a circle. "Radiator" No. 5 (33¾ sq ft) lies in section 3 and by using the column headed 7-8 in Table 2, *Section* 3, it will be evident that this "radiator" requires ¾-in. "radiator" branches and an 11/32-in. orifice. "Radiator" No. 4

TABLE 3. CAPACITIES OF SMALL RISERS AND RISER BRANCHES. (PIPE SIZES FOR THE BRANCHES THE SAME AS FOR THE RISERS)

PIPE SIZES		CAPACITY IN EQUIVALENT SQUARE FEET OF DIRECT RADIATION			PIPE SIZES	
SUPPLY	RETURN	SECTION 4	SECTION 3	SECTION 2	SECTION 1	SUPPLY	RETURN
¼	¼	14.6	23.6	31.6	36.9	½	½
½	½	19.9	31.9	42.5	49.6	½	¾
¾	¾	31.1	49.1	65.0	75.6	¾	¾
¾	1	40.7	64.2	85.0	99.0	¾	1
1	1	59.5	92.8	122.4	143.2	1	1
1	1¼	78.9	125.0	165.0	193.0	1	1¼
1¼	1¼	121.0	190.2	253.0	295.0	1¼	1¼
1¼	1½	149.2	237.6	304.0	354.0	1¼	1½
1½	1½	181.4	288.0	378.0	442.5	1½	1½

TABLE 4. CAPACITIES OF LARGE RISERS AND RISER BRANCHES. (PIPE SIZES FOR THE BRANCHES LARGER THAN FOR THE RISERS)

PIPE SIZES			CAPACITY IN EQUIVALENT SQUARE FEET OF DIRECT RADIATION				PIPE SIZES			
SUPPLY RISER	RETURN RISER	SUPPLY RISER BRANCH	RETURN RISER BRANCH	SECTION 4	SECTION 3	SECTION 2	SECTION 1	SUPPLY RISER	RETURN RISER	SUPPLY RISER BRANCH	RETURN RISER BRANCH
1¼	2	2	2	232	369	485	567	1¼	2	2	2
2	2	2	2½	356	568	744	872	2	2	2	2½
2	2½	2½	2½	424	676	885	1040	2	2½	2½	2½
2½	2½	3	3	577	914	1197	1400	2½	2½	3	3
2½	3	3	3	705	1123			2½	3	3	3
3	3	3½	4	1020				3	3	3½	4
3	3½	4	4	1169				3½	3½	4	4

lies almost exactly on the dividing line between section 3 and section 2. Since it is a small "radiator," it probably will be advisable to place it in section 3. This arrangement also distributes the "radiation" among the sections somewhat more uniformly than would be the case if it were placed in section 2. Accordingly, again using the column headed 7-8 in Table 2, *Section* 3, it will be found that 31½ sq ft require ¾-in. "radiator" branches and a 21/64-in. orifice. "Radiators" No. 3 and No. 2 lie in section 2, and by using the column headed 7-8 in *Section* 2 of Table 2, it will

be found that No. 3 (22½ sq ft) requires ½-in. branches and a ¼-in. orifice, while No. 2 (63 sq ft) requires ¾-in. branches and a 7/16-in. orifice. "Radiator" No. 1 lies in section 1, and by using the column headed 7-8 in *Section* 1 of Table 2, it is apparent that a 35-sq ft "radiator" requires ½-in. branches and a 5/16-in. orifice. The foregoing data may be expressed in compact tabular form as follows:

"Radiator" Number	Size of "Radiator" in Equivalent Square Feet	Section of Line	Size of "Radiator" Branches	Size of Drill Required to Bore Orifices
7	57	4	1	5/8
6	57	4	1	5/8
5	33¾	3	¾	11/32
4	31½	3	¾	21/64
3	22½	2	½	¼
2	63	2	¾	7/16
1	35	1	½	5/16

The right hand branch is 66½ ft long. Section 4 is therefore 9.5 ft long and each of the other sections, 19.0 ft long. This places "radiator" No. 10 in section 4, "radiator" No. 9 in section 2, and "radiator" No. 8 in section 1. The required sizes for orifices and "radiator" branches may again be found from Table 2 and tabulated as follows:

"Radiator" Number	Size of "Radiator" in Equivalent Square Feet	Section of Line	Size of "Radiator" Branches	Size of Drill Required to Bore Orifices
10	75	4	1¼	19/32
9	86¼	2	1	15/32
8	60	1	¾	3/8

Step 7.

Does not apply to a one-story installation.

Step 8.

The simple method of selecting sizes for mains is to apply the data in Table 5, and for each part of the line to select the sizes that have a capacity equal to or greater than the load which that particular part of the line is required to carry. Since the lowest portion of the mains is 3½ ft above the mid-point of the "boiler," the required data are to be found in the upper half of Table 5. The left hand branch is 92½ ft long, so the column headed 90-100 is used. Thus, the portion of the left hand branch lying between "radiators" No. 7 and No. 6 must carry 57 sq ft. The first number in the 90-100 column larger than 57 is 67. Following the line under 67 to either the right or the left margin, it will be found that the supply main must be 1¼ in. and the return main must also be 1¼ in. Proceeding in a similar manner, the following sizes for all portions of the mains are obtained:

"Radiators" Between Which Mains Lie	Required Capacity	Supply Size	Return Size
No. 7 and No. 6	57	1¼	1¼
No. 6 and No. 5	114	1½	2
No. 5 and No. 4	147¾	2	2
No. 4 and No. 3	179¼	2	2
No. 3 and No. 2	201¾	2	2½
No. 2 and No. 1	264¾	2½	2½
No. 1 and Boiler	299¾	2½	2½
No. 10 and No. 9	75	1¼	1¼
No. 9 and No. 8	161¼	2	2
No. 8 and Boiler	221¼	2	2

TABLE 5. CAPACITIES OF MAINS IN EQUIVALENT SQUARE FEET OF DIRECT RADIATION

DISTANCE FROM MID-POINT OF BOILER TO LOWEST POINT OF SUPPLY MAIN 3FT-4FT.

Pipe Sizes Supply	Pipe Sizes Return	Length of Line in Feet 0-20	20-25	25-30	30-35	35-40	40-45	45-50	50-60	60-70	70-80	80-90	90-100	100-120	120-140	140-160	160-180	180-200	200-240	240-280	280-330	Pipe Sizes Supply	Pipe Sizes Return
½"	½"	18	17	16	14	13	12	11	10	9	8	7	7	6	5							½"	½"
½"	¾"	24	23	21	19½	18	17	16	14	13	11	10	10	8	7							¾"	¾"
¾"	¾"	36	35	32	30	28	26	25	22	20	18	16	15	13	12	12	11	10	9	9	8	¾"	¾"
¾"	1"	49	45	42	39	38	34	32	28	26	23	22	20	18	17	15	14	13	12	11	10	¾"	1"
1"	1"	69	64	60	55	52	48	45	41	37	34	32	30	28	25	21	19	18	16	16	13	1"	1"
1"	1¼"	92	85	80	74	70	65	62	56	51	47	44	41	37	35	30	27	25	23	21	16	1"	1¼"
1¼"	1¼"	136	127	119	112	106	100	95	87	81	75	70	67	60	55	50	46	43	38	34	29	1¼"	1¼"
1¼"	1½"	162	151	141	133	125	119	114	104	97	90	85	81	73	67	61	58	54	48	42	36	1¼"	1½"
1½"	1½"	201	185	172	163	154	147	140	129	121	114	106	103	93	85	79	74	71	65	56	49	1½"	1½"
1½"	2"	262	246	230	213	202	192	184	166	157	147	140	133	122	112	106	100	98	84	76	64	1½"	2"
2"	2"	376	349	327	305	291	277	266	242	227	210	196	189	176	162	150	143	134	124	114	100	2"	2"
2"	2½"	460	423	392	365	349	332	319	295	279	261	250	236	219	201	186	175	165	148	137	120	2"	2½"
2½"	2½"	560	515	480	456	437	415	400	380	360	345	323	315	287	265	245	225	209	185	167	145	2½"	2½"
2½"	3"	700	650	611	580	557	531	510	485	455	433	412	391	357	335	311	292	278	241	218	197	2½"	3"
3"	3"	931	884	842	805	770	740	710	665	620	580	550	521	474	446	417	395	377	346	319	285	3"	3"
3"	3½"	1105	1046	994	946	910	874	840	787	737	693	659	626	575	536	504	479	457	417	383	342	3"	3½"
3½"	3½"	1359	1280	1215	1156	1111	1067	1030	965	912	865	825	788	731	685	645	610	580	526	480	428	3½"	3½"
3½"	4"	1521	1462	1392	1330	1274	1226	1177	1101	1040	989	947	912	853	803	761	723	686	619	559	488	3½"	4"
4"	4"	1805	1711	1635	1568	1495	1432	1375	1280	1210	1158	1115	1080	1025	977	931	889	845	756	673	570	4"	4"
4"	5"	2280	2170	2070	1990	1908	1821	1764	1630	1554	1481	1432	1387	1308	1242	1181	1115	1075	964	854	750	4"	5"
5"	5"	3090	2950	2830	2730	2620	2525	2445	2290	2170	2075	2000	1940	1806	1700	1605	1530	1460	1330	1210	1100	5"	5"
5"	6"	3720	3580	3450	3310	3200	3070	2980	2810	2680	2570	2480	2390	2250	2115	2005	1905	1820	1659	1526	1390	5"	6"
6"	6"	4660	4570	4460	4280	4110	3940	3800	3650	3510	3380	3265	3160	2970	2800	2660	2520	2420	2235	2080	1890	6"	6"

DISTANCE FROM MID-POINT OF HEATER TO LOWEST POINT OF SUPPLY MAIN 4 FT.-5 FT.

Pipe Sizes Supply	Pipe Sizes Return	Length of Line in Feet 0-20	20-25	25-30	30-35	35-40	40-45	45-50	50-60	60-70	70-80	80-90	90-100	100-120	120-140	140-160	160-180	180-200	200-240	240-280	280-330	Pipe Sizes Supply	Pipe Sizes Return
¼"	½"	21	20	18	16	15	14	13	11	10	9	9	8	7	6							¼"	½"
½"	¾"	28	26	24	22	21	20	19	17	15	14	12	11	9	8							½"	¾"
¾"	¾"	44	40	37	35	32	30	29	25	23	21	19	17	15	14	14	13	13	12	10	9	¾"	¾"
¾"	1"	57	52	49	45	42	39	37	33	30	27	25	23	21	19	17	16	15	14	13	12	¾"	1"
1"	1"	80	74	69	64	60	56	53	47	43	39	37	35	31	28	24	22	20	18	17	15	1"	1"
1"	1¼"	106	98	91	85	80	75	72	64	59	54	51	47	42	37	34	31	29	27	24	21	1"	1¼"
1¼"	1¼"	157	146	137	128	122	115	111	102	94	87	81	77	69	63	58	54	50	44	39	33	1¼"	1¼"
1¼"	1½"	187	174	163	154	144	137	132	120	112	104	98	93	83	76	71	66	62	55	49	42	1¼"	1½"
1½"	1½"	232	214	198	186	178	170	162	149	140	132	125	119	107	98	91	89	82	73	65	57	1½"	1½"
1½"	2"	303	280	261	245	233	222	212	195	181	175	162	153	141	129	120	113	110	97	87	74	1½"	2"
2"	2"	435	403	378	352	336	320	307	280	260	242	226	216	203	187	173	165	155	143	132	115	2"	2"
2"	2½"	524	480	450	421	403	384	368	342	322	301	285	272	250	230	215	202	188	171	156	139	2"	2½"
2½"	2½"	646	595	555	527	504	480	462	439	419	398	380	366	331	306	283	260	241	213	197	179	2½"	2½"
2½"	3"	808	750	705	670	643	614	590	558	526	500	476	452	417	384	358	332	310	278	257	238	2½"	3"
3"	3"	1075	1020	972	930	890	855	820	768	716	670	635	601	546	515	481	456	435	400	368	329	3"	3"
3"	3½"	1275	1210	1148	1090	1050	1009	970	908	850	800	760	725	664	621	582	554	527	481	443	395	3"	3½"
3½"	3½"	1568	1480	1403	1333	1262	1231	1190	1111	1050	998	955	910	845	788	745	705	670	607	555	494	3½"	3½"
3½"	4"	1760	1690	1610	1535	1470	1416	1360	1272	1200	1140	1093	1052	985	928	879	835	793	715	641	564	3½"	4"
4"	4"	2080	1978	1890	1810	1728	1668	1588	1480	1397	1337	1290	1250	1165	1130	1074	1025	975	873	776	659	4"	4"
4"	5"	2650	2500	2390	2300	2200	2110	2035	1880	1790	1710	1650	1600	1510	1433	1365	1288	1230	1111	986	866	4"	5"
5"	5"	3570	3400	3270	3150	3030	2920	2825	2640	2500	2395	2310	2225	2085	1963	1852	1760	1686	1535	1400	1270	5"	5"
5"	6"	4300	4140	4000	3840	3700	3570	3440	3240	3090	2970	2870	2780	2600	2440	2310	2200	2100	1914	1760	1605	5"	6"
6"	6"	5390	5260	5150	4950	4750	4550	4390	4210	4050	3900	3770	3650	3430	3235	3070	2910	2800	2580	2400	2180	6"	6"

Step 9.

Since there is only one elbow per main close to the "boiler" step 9 does not apply.

Step 10.

The sizes for the "boiler" uptakes and "boiler" returns are obtained from Table 6. Since the left branch carries 299¾ sq ft of "radiation," the required size is 2½-in. The right branch carries 221¼ sq ft and therefore the required size for it is also 2½-in.

In most cases it will be found convenient to combine all of the foregoing steps in one table as follows:

"Radiator" No.	Size of "Radiator" in Sq Ft	Section of Line	Size of "Radiator" Branches	Size of Drill Required to Bore Orifices	Required Capacity for Mains	Size of Supply Mains	Size of Return Mains	Size of "Boiler" Uptakes and Boiler Returns
7	57	4	1	5/8	57	1¼	1¼	
6	57	4	1	5/8	114	1½	2	
5	33¾	3	¾	11/32	147¾	2	2	
4	31½	3	¾	21/64	179¼	2	2	
3	22½	2	½	¼	201¾	2	2½	
2	63	2	¾	7/16	264¾	2½	2½	
1	35	1	½	9/16	299¾	2½	2½	
								2½
10	75	4	1¼	19/32	75	1¼	1¼	
9	86¼	2	1	15/32	161¼	2	2	
8	60	1	¾	3/8	221¼	2	2	
								2½

A MORE ACCURATE AND ECONOMICAL METHOD

In making an accurate design, steps 1 to 7 are the same as for the simple design but in step 8, by using a more elaborate method of selecting the sizes of the mains, a considerable saving can be made. It will at once be recognized that since the pipe sizes recommended in the 3 Ft—4 Ft part of Table 5 are satisfactory when the lowest portion of the mains is 3 ft above the mid-point of the heater, they must be larger than necessary when the distance above the mid-point is more than 3 ft. In order to aid in the selection of these smaller sizes Table 7 has been provided.

Example. Suppose that a line is actually 100 ft long but that the distance from the mid-point of the heater to the lowest portion of the mains is 3 ft 4 in. instead of an even 3 ft. On the 3 Ft-4 Ft side of Table 7 under the heading 3 *ft*-4 *in.* will be found the equivalent length factor 0.891. Consequently, the equivalent length of this 100-ft line is 0.891 × 100 ft or 89.1 ft and the sizes may be taken from the 80-90 column of Table 5 instead of from the 90-100 column of that table and a saving usually will result.

If a still further reduction in pipe sizes is desired, it can be obtained provided the designer is sufficiently careful. The capacities given in Table 5 are the maximums that can be carried without exceeding the available friction per foot, but whenever any part of the line is carrying less than its capacity load as determined by Table 5, its friction is less than the maximum allowable which means that it will be permissible for some other part of the mains to

carry more than its rated capacity, provided the extra friction caused by this excess is equal to or less than the friction saved where the load is less than the rated capacity. Since the friction in the mains varies approximately as the square of the load carried, it is very important that this part of the work be done carefully. The friction increase due to a given overload is much greater than the friction decrease due to an equal underload.

The general method for designing mains accurately may be divided into 4 steps as follows:

Step 1.

Find the equivalent length of the line by means of Table 7.

Step 2.

Make a tentative selection of pipe sizes from Table 5 using the equivalent length of the line and selecting the pipe sizes having the capacity nearest to the required capacity. When in doubt as to which of two sizes is nearer, choose the larger.

Step 3.

Calculate the friction factors for the various parts of the mains, making a separate calculation for each part that carries a different load or has different pipe sizes from the adjacent parts. The friction factor is calculated by means of the following formula:

$$F = L - \frac{LS^2}{C^2}$$

where

F = the friction factor.

L = the length, including elbow equivalents, of the part of the mains being checked. If the supply and return mains are not the same length, or do not contain the same number of elbows, take the average in each case. The equivalent of an elbow in feet of pipe may be taken as twice its nominal diameter in inches.

C = the capacity of the mains as given in Table 5.

S = the actual load the mains must carry in equivalent square feet.

Step 4.

Arrange the positive factors in one column and the negative factors in another. If the sum of the positive factors is equal to or greater than the sum of the negative factors, the design is satisfactory. If the sum of the negative factors is larger, some of the pipe sizes must be increased. Often it is possible to effect a considerable saving by being careful in selecting the places where the mains are to be overloaded.

In applying these methods to the left branch of the one-story layout of Fig. 10, the steps are as follows:

Step 1.

Since the lowest portion of the mains is 3 ft-6 in. above the mid-point of the heater, the equivalent length factor is found in Table 7 under the heading 3 *ft-6 in.* This factor is 0.845 and since the actual length of the line was previously found to be 92½ ft, the equivalent length is 0.845 × 92.5 or 78.2 ft. Therefore, the pipe sizes are to be taken from the 70-80 column of the 3 *Ft-4 Ft* part of Table 5.

Steps 2, 3 and 4

These three steps are best arranged as follows:

1	2	3	4	5	6	7	8	9
"Radiators" Between Which Mains Lie	Load (S)	Pipe Sizes		Length of Mains (L)	Rated Capacity (C)	$\frac{LS^2}{C^2}$	Friction Factors $L - \frac{LS^2}{C^2}$	
		Supply	Return				Positive	Negative
7-6	57	1¼	1½	9½	75	5.48	4.02	....
6-5	114	1½	1½	31½	114	....	0	0
5-4	147¾	1½	2	12½	147	12.63	....	0.13
4-3	179¼	2	2	7¼	210	5.28	1.97	....
3-2	201¾	2	2	18½	210	17.05	1.45	....
2-1	264¾	2	2½	14¼	261	14.63	....	0.38
1-Boiler	299¾	2	2½	16½	261	21.75	....	5.25
						Totals	7.44	5.76

Several points regarding the foregoing tabulation should be noted. The *first* is that in selecting the pipe sizes given in columns 3 and 4 the mains nearest the heater were overloaded, that is, the actual load on this part of the mains is greater than their rated capacities as given in Table 5. This can nearly always be done and usually will save money because it reduces the quantity of large pipe needed.

The *second* relates to column 5. Since the equivalent of an elbow in feet of pipe is twice its nominal diameter in inches, the average equivalent for two mains, each of which contains an elbow, will be simply the sum of the nominal diameters of the two elbows. The numbers in the column in question were obtained by simply adding to the average length of the mains the nominal diameter of each elbow. Thus, the first three numbers in this column were obtained as follows:

$$7 + 1\tfrac{1}{4} + 1\tfrac{1}{4} = 9\tfrac{1}{2}$$
$$28\tfrac{1}{2} + 1\tfrac{1}{2} + 1\tfrac{1}{2} = 31\tfrac{1}{2}$$
$$9 + 1\tfrac{1}{2} + 2 = 12\tfrac{1}{2}$$

The *third* point to note is that the calculations have been so arranged that the numbers in each column can be obtained from the numbers in other columns either by a single slide rule operation or by a single subtraction.

The tabulated calculations for the more accurate design of the right branch of Fig. 10 are as follows:

$66.5 \times 0.845 = 56.2$ EQUIVALENT LENGTH OF BRANCH

"Radiators" Between Which Mains Lie	Load (S)	Pipe Sizes		Length of Mains (L)	Rated Capacity (C)	$\frac{LS^2}{C^2}$	Friction Factors $L - \frac{LS^2}{C^2}$	
		Supply	Return				Positive	Negative
10-9	75	1¼	1¼	31½	87	23.40	8.10	....
9-8	161¼	1½	2	22½	166	21.25	1.25	....
8-Boiler	221¼	1½	2	12½	166	21.75	....	9.250
							9.35	9.250

Steps 9 and 10 are the same as for the simple method.

The pipe sizes for the mains that were obtained by the more accurate method are shown in squares on Fig. 10.

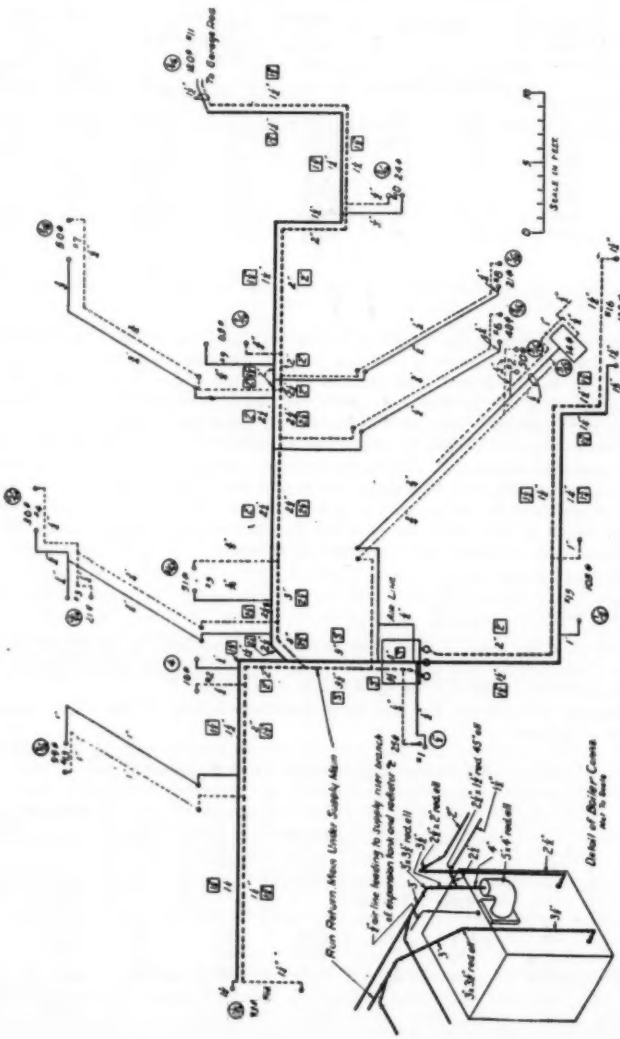


FIG. 11. LAYOUT FOR A TWO-STORY RESIDENCE WITH BRANCHED MAINS

BRANCHED LINE IN A TWO-STORY RESIDENCE

In Fig. 11 is shown the layout of a two-story plant that has a branched line. In making the layout it has been assumed that the return main is to be run directly under the supply main, since this usually is the simplest and cheapest method of handling a branch. The "radiators" are to have the high supply connection of Figs. 4*B* and 5*B*. The riser branches are to be according to Fig. 6*A* and the boiler is to be connected according to Fig. 8*C*.

In designing a system of this type it is necessary to begin with the longer branch. This branch is divided into sections just as though the shorter branch did not exist. Thus, since it is 65 ft from the top of the "boiler" uptake to the connections of radiator No. 11, the lengths of the sections are as follows:

Section 4: $\frac{65}{7} = 9.3$ ft

Sections 3, 2 and 1: $65 \times 2/7 = 18.6$ ft

To obtain the lengths of the sections in the shorter branch, measure the lengths of both branches beginning at the junction and multiply the lengths of the sections in the longer branch by the ratio of the length of the shorter branch to the length of the longer branch. Thus, the distance along the mains from the junction to "radiator" No. 14 is 26 ft and the distance from the junction to "radiator" No. 11 is 55 ft. Consequently, the lengths of the sections in the shorter branch are:

Section 4: $9.3 \times 26/55 = 4.4$ ft

Sections 3, 2 and 1: $18.6 \times 26/55 = 8.8$ ft

Total length (for purpose of selecting sizes of mains): $65 \times 26/55 = 30.7$ ft.

The short line at the bottom of the layout is designed just the same as for a one-story system.

In tabulating the calculations for a two-story installation, an extra column giving the height of the "radiator" above the mid-point of the "boiler" is needed and also columns for riser sizes which are sometimes larger than the "radiator" branches of second floor "radiators." The tabulated calculations are shown in Table 8.

The method of obtaining the numbers in the first column of the tabulation below, particularly the 30.7, has already been explained. The numbers in the second column correspond to the "radiator" numbers given on the layout. The numbers in the third column give the section in which each "radiator" lies and the fourth column gives the vertical distances from the mid-point of the "boiler" to the "radiator." Since the high supply connection is used this distance is measured to the middle of the "radiators." There is a slight variation because some of the "radiators" are taller than others.

The fifth column headed *Square Feet for "Radiator" Branches* contains simply the rating of each "radiator" in equivalent square feet. The sixth column headed *Square Feet for Risers and Riser Branches* gives the load that must be carried by each riser. In case the riser carries more than one "radiator," the ratings must be added as in the case of "radiators" No. 3 and No. 4.

The seventh column headed *Square Feet for Mains* contains the load that each part of the mains must carry. These loads are obtained by adding successively the loads contributed by each pair of riser or "radiator" branches

TABLE 6—CAPACITIES OF BOILER UPTAKES AND BOILER RETURNS

PIPE SIZES	½	¾	1	1¼	1½	2	2½	3	3½	4	5	6	7	8	PIPE SIZES
CAPACITY IN EQUIVALENT SQ FT.	13	26	49	86	122	216	353	542	713	940	1520	2360	3220	4200	CAPACITY IN EQUIVALENT SQ FT.

TABLE 7. EQUIVALENT LENGTH FACTORS

USE WITH 3FT-4FT. TABLE							ACTUAL DISTANCE ABOVE MIDPOINT	USE WITH 4FT-5FT. TABLE							
3'-0"	3'-2"	3'-4"	3'-6"	3'-8"	3'-10"	4'-0"	FACTOR	4'-0"	4'-2"	4'-4"	4'-6"	4'-8"	4'-10"	5'-0"	5'-2"	5'-4"	5'-6"
1.000	0.943	0.891	0.845	0.802	0.765	0.730		1.000	0.955	0.915	0.878	0.845	0.812	0.782	0.754	0.730	0.707

TABLE 8. VERTICAL DISTANCE FROM MID-POINT OF BOILER TO LOWEST POINT IN SUPPLY MAIN 3½ FT

Length of Main Branch In Feet	Number Radiator	Section of Line	Height Above Mid-Point of "Boiler"	Square Feet for "Radiator" Branches	Square Feet for Risers and Riser Branches	Square Feet for Mains	Table 2		Table 3		Table 5		Table 6	
									Risers and Riser Branches		3 Ft.-4 Ft. Mains			
							Drill Size for Orifice	"Radiator" Branches	Supply	Return	Supply	Return	Boiler Uptake	Boiler Return
65	11	4	8' 0"	120	...	120	¾	1½	...	...	1½	1½	...	...
	10	3	7' 4"	24	...	144	5/16	½	...	...	1½	2	...	...
	9	2	8' 0"	65	...	209	7/16	¾	...	...	2	2	...	...
	8	2	17' 0"	21	21	230	7/32	½	½	½	2	2½	...	...
	7	2	17' 0"	50	50	280	⅜	½	¾	¾	2½	2½	...	...
	6	2	17' 0"	40	40	320	5/16	½	½	¾	2½	2½	...	...
	5	1	7' 6½"	51	...	371	21/64	¾	...	...	2½	3	...	...
	4	1	16' 6½"	36	...	...	17/64	½	...	...	...	...	...	...
	3	1	17' 0"	21	57	428	13/64	½	¾	¾	2½	3	...	...
30.7	14	4	7' 6"	93	...	93	11/16	1¼	...	...	1¼	1¼	...	...
	13	2	17' 0"	99	99	192	7/16	1	1	1	1½	2	...	...
	12	2	7' 6"	16	...	208	4	½	...	...	1½	2	...	...
			Junction of Branches											
	..	..	....	...	...	636	....	...	...	...	3	3½	...	...
65	Exp.	1	26' 0"	14	14	...	5/32	½	½	½	...	...	...	...
	2	1	17' 0"	50	64	700	5/16	½	¾	¾	3	3½	...	...
	1	1	8' 4½"	25	...	725	15/64	½	...	...	3	3½	...	4
30	16	4	7' 8½"	125	...	125	13/16	1½	...	...	1¾	1½	...	...
	15	2	7' 6½"	105	...	230	½	1	...	...	1½	2	...	2½
	..	..	.....	...	...	955	...	...	...	...	...	...	5	...

where they enter the mains. The remaining columns containing orifice and pipe sizes are obtained by means of the tables. At the head of each column is indicated the table which applies to that particular column.

It will be noted that the supply connection to "radiator" No. 16 is larger than the main that it joins. This is due to the shortness of the line and does no harm, but it could have been avoided as will be explained in the answer to question 19.

ECONOMICAL METHOD APPLIED TO A BRANCH LINE

Since it is usually most economical to overload a system close to the "boiler," it will be advisable in the case of a branched line to calculate the part between the junction and the "boiler" first, choosing the pipe sizes so as to make the total friction factor negative and then to so choose the pipe sizes for each branch that each shall have a total positive friction factor at least as large as the total negative friction factor of the part between the junction and the "boiler." The lowest part of the supply main is 3½ ft above the mid-point of the heater so the equivalent length factor according to Table 7 is 0.845.

The tabulated calculations are as follows:

$65 \times 0.845 = 54.9$, equivalent length of 65-ft line.
$30.7 \times 0.845 = 25.9$, equivalent length of 30.7-ft line.
$30 \times 0.845 = 25.4$, equivalent length of 30-ft line.

The calculations for the accompanying tabulation are so similar to the accurate main calculations, for the one-story installation that they can probably be

"Radiators" Between Which Mains Lie	Load (S)	Pipe Sizes		Length of Mains (L)	Rated Capacity (C)	$\frac{LS^2}{C^2}$	Friction Factors $L - \frac{LS^2}{C^2}$	
		Supply	Return				Positive	Negative
Junc.-2	636	3	3	6.0	665	5.48	0.52	....
2-1	686	3	3	2.5	665	2.66	....	0.16
1-Boiler	711	3	3	3.5	665	4.00	....	0.50
							0.52	0.66
						Total		0.14
11-10	120	1½	1½	20.0	129	17.32	2.68	....
10-9	144	1½	2	18.0	166	13.56	4.44	....
9-8	209	1½	2	1.5	166	2.38	....	0.88
8-7	230	2	2	1.5	242	1.36	0.14	....
7-6	280	2	2½	3.5	296	4.09	....	0.59
6-5	320	2	2½	10.0	296	11.70	....	1.70
5-4 & 3	371	2½	2½	3.5	380	3.34	0.16	....
4 & 3-Junc.	428	2½	2½	5.5	380	6.99	....	1.49
							7.42	4.76
14-13	93	1¼	1¼	14.0	119	8.56	5.44	....
13-12	192	1½	1½	8.0	172	9.97	....	1.97
12-Junc.	208	1½	2	7.5	230	9.30	....	1.86
							5.44	3.38

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TABLE 9. CALCULATIONS FOR A THREE-STORY INSTALLATION

Radiator Number	(*L*) Distances Between Branches Including Elbow Equivalents	Section of Line	Height Above Mid Point of Boiler	Square Feet for Radiator Branches	Square Feet for Risers and Riser Branches	(*S*) Square Feet for Mains	Table 2		Table 3 Risers and Riser Branches		Table 5 (3 Ft.-4 Ft.) (90-100) (Column) Mains		Table 6 Boiler		(*C*) Rated Capacity of Mains	$\frac{LS^2}{C^2}$	$L - \frac{LS^2}{C^2}$	
							Drill Size for Orifice	"Radiator" Branches	Supply	Return	Supply	Return	Uptake	Return			Friction Plus	Factors Minus
35	..	4	25′ 5″	15	15	...	7/32	1/2	1/2	1/2	...	...	...	...	...	...	...	...
25	..	4	16′ 5″	13¾	28¾	...	19/64	1/2	3/4	3/4	...	...	...	...	...	...	...	...
15	..	4	7′ 6″	13¾	...	...	9/32	1/2	...	...	...	...	...	...	...	...	...	...
..	13	..	....	...	...	42½	...	...	...	...	1	1¼	...	...	47	10.63	2.37	...
34	..	4	25′ 5″	15	15	...	7/32	1/2	1/2	1/2	...	...	...	...	...	...	...	...
24	..	4	16′ 5″	74¼	89¼	...	9/16	1	1¼	1¼	...	...	...	...	...	...	...	...
14	..	4	7′ 6″	74¼	...	...	19/32	1¼	...	...	...	...	...	...	...	...	...	...
..	24	..	....	...	...	206	...	...	...	...	2	2	...	...	218	21.45	2.55	...
33	..	3	25′ 5″	75	75	...	13/32	3/4	1	1	...	...	...	...	...	...	...	...
23	..	3	16′ 5″	74¼	149¼	...	1/2	3/4	1¼	1¼	...	...	...	...	...	...	...	...
13	..	3	7′ 6″	13¾	...	...	7/32	1/2	...	...	...	...	...	...	...	...	...	...
..	31	..	....	...	...	369	...	...	...	...	2½	2½	...	...	361	32.40	...	1.40
32	..	2	25′ 5″	55	55	...	5/16	1/2	3/4	3/4	...	...	...	...	...	...	...	...
22	..	2	16′ 5″	38½	93½	...	9/32	1/2	1	1	...	...	...	...	...	...	...	...
12	..	2	7′ 6″	38½	...	...	5/16	3/4	...	...	...	...	...	...	...	...	...	...
..	26	..	....	...	...	501	...	...	...	...	3	3	...	...	601	18.05	7.95	...
31	..	1	25′ 5″	75	75	...	3/8	1/2	3/4	3/4	...	...	...	...	...	...	...	...
21	..	1	16′ 5″	74¼	149¼	...	11/32	3/4	1	1¼	...	...	...	...	...	...	...	...
11	..	1	7′ 6″	74¼	...	...	7/16	3/4	...	...	...	...	...	...	...	...	...	...
..	14	..	....	...	...	724½	...	...	...	...	3	3	...	...	601	20.30	...	6.30
Boiler	..	..	....	...	...	...	...	...	...	...	...	...	4	4	...Totals		12.87	7.70

readily understood without comments. No attempt was made to make a more accurate design for the short line at the bottom of the figure because the design was evidently already as close as it was practical to endeavor to attain.

ECONOMICAL METHOD APPLIED TO A BRANCH OF THREE-STORY INSTALLATION

The calculations for a three-story installation will be illustrated by means of the experimental plant shown in Fig. 1. In this case, since the lowest point on the supply main is exactly 4 ft above the mid-point of the "boiler," there is no need to use Table 7. Also, for the same reason the lower half of Table 5 is used instead of the upper half. The friction factor calculations have been combined with the other calculations, all of which are given in Table 9.

METHODS USED IN CALCULATING

In making Table 5 giving the capacities of mains, the pressure head available at a point 3 ft above the mid-point of the heater was first calculated. Ten per cent of this was set apart to overcome the friction in the "boiler" fittings, including the uptake and return, and the elbows that connect them to the mains. Fifteen per cent more was set apart to be available at the end of the line to make certain that the last "radiator" started in the right direction. The remaining 75 per cent was reserved for the mains themselves. Allowance was made for four elbows in each main. The "boiler" temperature differential was taken as 40 deg in order to obtain the available pressure head, which was 174 x 3 or 522 mil-inches. To get the friction in the mains, the quantity of water flowing through them was taken as the quantity that would be required to flow through the "radiators" at an average temperature differential of 28.5 deg when they were dissipating heat at their full rated capacity.

After the capacities of the mains having the same size supply and return were calculated, the capacities of the mixed sizes were calculated by the formula:

$$C = \frac{1.905\, C_1\, C_2}{C_1 - C_2}$$

where

C = the capacity of the mixed size.

C_1 = the capacity of a line having both pipes the same size as the supply main of the mixed line.

C_2 = the capacity of a line having both pipes the same size as the return main of the mixed line.

When the capacity of one of the sizes is double that of the other, formula (1) is accurate to about 0.2 per cent and it is sufficiently accurate for practical purposes over a considerably wider range than any for which it is likely to be used.

In order to obtain the pressure heads available for the various "radiator" and riser branches the average pressure between the mains at a height of 4 ft above the heater was computed for each section of the line. The pressure heads for the first floor "radiator" branches were then computed by adding the head generated in the "radiator" circuit to the head present in that particular part

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of the mains. The "radiators" in the first section of a line were to run at a temperature differential of 37.5 deg, those in the second section at 34.5 deg, those in the third section at 29.5 deg, and those in the fourth section at 22.5 deg.

After the available pressure heads had been determined, the capacities of the orifices and "radiator" branches for the first floor "radiators" were determined by the formula:

$$S = T\sqrt{\frac{PH}{1 - KP}} \tag{2}$$

where

S = the capacity in equivalent square feet
T = the temperature differential in degrees Fahrenheit
H = the available pressure head in milinches of water
K = the friction head in milinches of water that would exist in the pipes and fittings, exclusive of the orifice, if they were connected to one equivalent square foot of heating surface operating at a temperature differential of one degree.

$$P = \frac{0.334\,(D - 0.046)^4\,d^4}{d^4 - (D - 0.046)^4} \tag{3}$$

where

D = the diameter of the orifice in inches
d = the actual internal diameter of the pipe in inches

From a hydrodynamic standpoint, formula (3) is not accurate because it is designed not simply to give the quantity of water which must flow through the "radiator" per second in order to produce a given friction head, but to admit a larger quantity of water to a small "radiator" or one high above the "boiler" and thus to compensate for the greater cooling in pipes that are unusually small or long. From the data given in the first part of this paper, it can be seen that it accomplished this purpose very well in the case of the experimental plant.

The risers in the fourth section have a maximum allowable friction of 16 milinches of water per vertical foot or 8 milinches per foot in each pipe when both pipes are the same size. In the third section the riser friction is 22½ milinches per vertical foot. In the second section it is 28 milinches per vertical foot, and in the first section it is 32 milinches per vertical foot. As in the case of the mains, the capacities for pipes of equal size were computed first and the mixed sizes calculated by means of the formula:

$$\frac{1.905\,C_1\,C_2}{C_1 - C_2}$$

The friction in the riser branches is held below an allowable maximum of 280 milinches in section 4, the sizes of one or both of the branches being made larger than the sizes of the corresponding risers whenever necessary. In the other sections the allowable maximum friction is greater but the general method is the same. The pressure head across an upper floor "radiator" branch cannot be determined as accurately as the pressure head across a first floor "radiator" branch because of the variable character of the riser friction. Mean values were therefore taken and the calculations made by means of formula (2).

Also, since one of the primary objects of these tables is to obtain costs as low as is consistent with reliability, a great deal of attention had to be given to safety factors.

QUESTIONS AND ANSWERS

Question 1: Why are ceiling "radiators" recommended for use in basements whenever basement "radiators" are required?

Answer: Principally because ceiling "radiators" which lie entirely above the mains can be listed as an integral part of Table 2, while some other type of "radiator" would require special tables.

Question 2: Is it not a disadvantage to have tables that require so many different sizes of drills?

Answer: It is, but one means of reducing the safety factors required is to figure orifice capacities in small steps. At any rate it probably will never be necessary to buy very many drills at one time because the average installation requires only from six to twelve different sizes.

Question 3: Is it worth while to go to all the trouble of making the more accurate design for the mains?

Answer: It is always worth while to find the equivalent length of a line. Whether it is worth while to figure the friction factors is questionable. It may be that for small buildings it will be more satisfactory to be careful to err on the side of safety when selecting the sizes for the mains. The capacities of the mains may be exceeded in places but these excesses must be small, much smaller than the underloads that are intended to off-set them. Also, the length of the line affected should be less for the overloaded than for the underloaded portions. However, the friction factor method should not be discarded without giving it an extended trial. It is much more exact than pure guessing and when a slide rule is used, the procedure is comparatively simple.

Question 4: How often may trouble be expected with systems designed according to these tables?

Answer: There is every reason to expect that circulation troubles will be almost unknown except in cases where a serious mistake has been made. However, no practical tests have been made on systems where some of the "radiators" are more than 30 ft above the mid-point of the "boiler" and it is always possible that an untried portion of a table may not work out exactly as expected.

Question 5: What should be done in case trouble does develop?

Answer: It is usually necessary to place smaller orifices in the "radiators" that are functioning satisfactorily. Sometimes covering the supply piping will improve conditions, but covering the return mains or branches is likely to make matters worse. Sometimes it is advisable to take the orifice out of the "radiator" that is not functioning properly but the installation of smaller orifices in the hot "radiators" is the most reliable remedy.

Question 6: In case a carefully designed system fails to circulate properly, what is the most probable cause?

Answer: Probably some of the orifices have been omitted. The omission of some of the orifices is almost certain to cause trouble and the omission of only one may sometimes result in serious consequences.

Question 7: Will the "radiators" on the end of a line get as hot as those near the "boiler"?

Answer: In a fairly tall building it is expected that the "radiators" in section 4 will be somewhat hotter than those in sections 1 and 2, while in low rambling buildings the "radiators" in the sections close to the "boiler" are likely to become somewhat hotter than those in section 4.

Question 8: Why was section 4 made shorter than the other sections?

Answer: Because, in an average system, there is a tendency for heating surface to be *bunched* at the end of a line. Since section 4 "radiators" require more water

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than similar "radiators" in other sections, too much heating surface in section 4 might overload the mains and cause trouble. This is especially true where the mains branch.

Question 9: Can these tables be extended so as to apply to taller buildings?

Answer: It seems probable that they can. So far as the writer has been able to ascertain, the only limitation upon the height to which hot water can be successfully used for heating is the limitation due to the hydrostatic pressure caused by a high column of water, provided proper tables have been developed.

Question 10: Can these tables be used to design systems having welded mains?

Answer: These tables can be used to design systems having welded mains provided the elbows in the mains are of the long sweep type and the riser branches and "radiator" branches are screwed. If welded riser and "radiator" branches are used there is some likelihood that it may be necessary to reduce orifice sizes in order to compensate for the fact that the long sweep welded elbows have lower friction than the screwed elbows.

Question 11: In reference to covered mains, would it seriously affect the circulation or the balance of a system if the supply pipes only were covered for one-story jobs and supply piping and return risers only for two- or three-story jobs?

Answer: On a line where most of the "radiators" are small, covering the supply piping would be likely to improve the circulation. On a multiple-story line having mostly large "radiators," it would probably cause the fourth section "radiators" to have a slightly higher average temperature than the remainder of the system. The average temperature of the system would, however, be higher and the temperature differentials lower than would be the case were no covering used. To sum up, covering all the supply piping and return risers, if any, would probably improve some systems and would never have any noticeable adverse effect provided the return mains and riser branches were left bare.

Question 12: If it is necessary to locate a small "radiator" on the end of a line, is there anything that can be done in designing the system to insure satisfactory operation?

Answer: There are certain precautions that can be taken in advance and an easy remedy that can be applied later if necessary. They are as follows:

1. Be sure that the mains, especially the return main, are large enough throughout the entire line.
2. In the doubtful "radiator," use an orifice two or three sizes larger than called for by the tables.
3. If necessary, cover all the supply piping leading to the "radiator" including both branches and risers, if any, and also cover the portions of the supply main where the pipe is small.

Question 13: Why does a closed system under pressure operate at a higher temperature differential than an open system?

Answer: If the open and closed systems were operated at the same temperatures there would be no appreciable difference in their temperature differentials, but when the closed system is raised to a temperature higher than is ever used for an open system, it dissipates heat more rapidly than does the open system and consequently its temperature differentials are higher.

Question 14: In the case of the tests on the experimental plant, why were the hottest and coolest "radiators" different when the system was closed than when it was open?

Answer: This was probably due to the fact that the "radiators" stood on a scaffold in one large room, with the top floor "radiators" very close to the ceiling. Under these conditions there was a tendency for the air surrounding the top floor "radiators" to be too hot, which, in turn, caused the top floor "radiators" to come to a higher temperature than they otherwise would have. This tendency was much

more pronounced with the closed system because the high temperature "radiators" heated the air to a higher temperature before it rose to the ceiling. Also, when the system was operated at high temperatures there was a tendency for the basement to overheat. The combined effect of these two tendencies was to raise the average temperatures of the "radiators" on the third floor and in the fourth section of the line and, as a result, the hottest and coolest "radiators" were not the same ones that had been hottest and coolest when the system was operated at lower temperatures. In addition to this, the steam pressure on the converter could not be held constant and consequently the temperature of the water leaving the converter was somewhat variable. As a result, a difference of two or three degrees between the temperatures of any of the "radiators" had very little if any significance.

Question 15: Referring to Fig. 4, why may "radiators" connected according to *A, D, G,* and *J,* be expected to give somewhat better circulation than "radiators" connected by the other methods illustrated?

Answer: When a "radiator" is connected according to *C, F, I,* or *L,* there is a tendency for eddies to be set up within the "radiator" so that the water goes up through some of the tubes and down through others. As a result, the incoming hot water is mixed with partly cooled water and a lower average "radiator" temperature results. This effect is most pronounced in the "radiators" close to the "boiler" where the temperature differences are large. When a "radiator" is connected according to *B, E, H,* or *K,* the velocity head of the entering water is frequently sufficient to carry the water entirely across the top of the "radiator" and part way down the return end before its energy is exhausted. As a result, the bottom corner of the "radiator" on the supply end is slow to heat. This condition is most noticeable in the fourth section and upper floor radiators where the circulation is rapid. In the connections shown at *A, D, G,* and *J,* the velocity head of the entering water is mostly used to overcome the greater friction encountered by the water that takes the longest possible path through the "radiator" so that the "radiator" temperature is likely to be more uniform than in either of the other cases. Also, when the high supply connection is used there is likely to be a smaller difference in temperature between the hottest "radiator" and the coolest because the heads generated in the "radiator" circuits are larger which tends to overcome the unavoidable differences in pressure that exist between the mains in different parts of the same section.

Question 16: Why should the measurement of the vertical distance above the mid-point of the heater be made to the middle of the radiator when the high supply connection is used and to the height of the unions when the low supply connection is used?

Answer: Because when the high supply connection is used the water in the "radiator" being cooler (on the average) and therefore heavier than the water in the top of the supply "radiator" branch creates a pressure head which should be taken into account when the size of the orifice is selected. When the low supply connection is used, the water in the "radiator" presses equally on both the supply and return "radiator" branches and consequently the water in the "radiator" cannot add anything to the available pressure head no matter how tall the "radiator" may be.

Question 17: Did the experimental plant give more air-binding trouble when a high pressure was used?

Answer: Yes. There was no trouble at all when the expansion tank was open but when it was closed "radiator" No. 34 became air-bound. A close examination of this radiator revealed a small previously unnoticed leak. Apparently when the system was cold and under a subatmospheric pressure, this leak had admitted the air which blocked the circulation to this "radiator" the next time the system was put in operation. It should be stated, however, that the experimental plant was never operated continuously for more than a few hours at a time and for that reason, no very reliable conclusions as to air-binding can be drawn at present.

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Question 18: In step 5 under *General Procedure in Designing a System* is it necessary to reduce elbows to equivalent feet of pipe before proceeding to divide the mains into sections?

Answer: No, it is not necessary to take account of elbows in sectioning the mains, but if a branch occurs close to the dividing line between two sections the locations of the elbows in the mains may help the designer to decide in which section that particular branch had better be placed.

Question 19: In the case of "radiator" No. 16 in Fig. 11, the supply "radiator" branch is larger than the supply main which feeds it. Could the size of this branch be safely reduced to the size of the main?

Answer: Yes. If the orifice size is increased to ⅞-in., the circulation will probably be just as good as if the tables had been followed exactly. The ⅞-in. orifice with the 1¼-in. supply branch and the 1½-in. return branch will give almost exactly the same friction as a 1 3/16-in. orifice with both branches 1½-in.

Question 20: If the size of the "boiler" uptake or "boiler" return called for by the tables is larger than the corresponding "boiler" opening, what should be done?

Answer: This is not likely to happen, but if it should, the small "boiler" connections can be compensated for by using larger mains than called for by the tables. There are so many fittings in the "boiler" connections that the mains will need to be enlarged for a considerable distance in order to fully compensate for an uptake or "boiler" return that is too small.

Question 21: Should the expansion pipe and the circulating pipe always be ½-in. and the overflow always 1-in., as shown in Fig. 9*A* and Fig. 9*B*?

Answer: Yes. According to Table 2, a pair of ½-in. risers can take care of 36.9 equivalent square feet. Since there are usually fewer elbows in the connections to an expansion tank than in the connections to a "radiator" and also since it is always desirable that the expansion tank dissipate as little heat as possible, these sizes should be ample.

Question 22: Is it necessary to put a check valve in the vent pipe when the expansion tank connections are made exactly according to Fig. 9*A* or Fig. 9*B*?

Answer: If either Fig. 9*A* or Fig. 9*B* is followed exactly, experience with the experimental plant indicates that a check valve will be unnecessary. When the cold water feed was wide open and the system full of water the reducing tee at the top of the overflow acted like an aspirator and air was drawn down through the vent pipe. Upon another occasion when the average temperature of the system was about 220 F, the valve between the expansion tank and the vent pipe was opened wide, but nothing came out of the vent except an almost imperceptible amount of flash steam. In case it should be necessary to run the upper portion of the overflow horizontally for a considerable distance, a check valve might be required.

Question 23: In case the expansion tank is connected to risers and requires an orifice, how should the size of the orifice be determined?

Answer: There is considerable latitude in determining the size of an expansion tank orifice and it need not be done accurately. A simple and satisfactory method is to select the orifice that will take care of a "radiator" the equivalent feet of which are equal to the actual external area of the tank.

CONCLUSIONS

The experimental results reported in this paper appear to be very satisfactory and there is sufficient reason to believe that every radiator on an open system designed according to the tables given can be counted on to deliver 150 Btu per hour per equivalent square foot without approaching the boiling point. Also, it seems safe to assume that closed systems designed with only 10 per

cent more heating surface than is required for steam will function satisfactorily provided the safety valve is set to blow off when the pressure at the top of the highest radiator is 15 or 20 lb per square inch, gage.

It is not known whether these pipe sizes are the most economical that can be incorporated in practical tables, but it is believed that they are fairly close to that ideal. In considering the economy of pipe sizes it should be remembered that one of the prime requisites of an economical installation is that there shall be no expensive changes or adjustments to make after the system is once installed.

In conclusion, acknowledgment is made to H. M. Hart and R. W. Shields for their help in putting this paper into the form that would be most useful to the industry, and also to Professors Crawford and Brewer of the Mechanical Engineering Department of Agricultural and Mechanical College of Texas for their kindness in donating the use of their laboratory and tools.

DISCUSSION

M. A. BERNHARDT (WRITTEN): If some comparatively simple rules or formulae could be found by which reasonable accuracy could be obtained in designing hot water heating systems, I believe the architects specifying hot water heat would have much to be thankful for.

It may be argued that a hot water heating system should be designed by a heating engineer and, admitting this for the moment, the architect should have available such information that would make it comparatively simple for him to check the engineer's design. Few residences employing hot water heat are designed by engineers and when they are designed by them, no two agree sufficiently to trust either one. Each engineer has his own pet theories about hot water heating and each will admit that there is no standard by which to measure the accuracy of his design and that the design is based largely on experience. The scheme worked before, therefore it should work now; why, he knows not.

As the architect for several Philadelphia suburban homes I chose hot water for heating, because in this city and its suburbs hot water heat is popular and it is extremely difficult to make people believe that any other system is as good. The systems were installed in the *usual* manner but they did not perform normally. We started to investigate, three or four engineers were called in and by the time each one rendered his opinion I came to the conclusion that each of them knew very little and I nothing about hot water heating. At this time I received the announcement of the publication of the new edition of your book on heating and ventilating and I quickly requested a copy but to my regret I found the chapter on hot water heating of no value in trying to solve my problem and so I returned the book with my comments.

In the average suburban home employing gravity hot water heat, the boiler is located in the basement and the mains and returns are, of course, also located there, and almost all the systems now used are of the closed type. The problems that always confront us in designing such a system are: (1) size of mains and risers; (2) method of taking off risers from main, that is, should connections be taken off the top—45 deg or the side, and should these take-offs be varied near the boiler or should they be kept uniform throughout; (3) should the take-offs for the first floor be the same as for the second floor; (4) even distribution of heat throughout the house; (5) effect of recessing rads in walls and putting grille over recess; (6) use of the new style fin type radiation for hot water; (7) if fin type radiation is used should piping be altered to suit; (8) use of orifices in risers; (9) does reducing of

size of mains and risers retard or accelerate circulation, and (10) economical temperature of water under which system should be operated to produce good results.

(1) The question of determining the size of mains and risers is always a serious one. Each engineer has a method of his own but it appears that Mr. Smith has determined a method that should be accurate. However, it is too complex for general use and if it could in some way be simplified, I am certain it would prove more helpful.

(2 and 3) There appears to be as many different theories about taking off risers from mains as there are ways in which it can be done. Each theory has its supporters but there is no real authority on the subject. It appears that many claim that the first floor rads should be taken off the top and the upper floors off the side. Some argue that this arrangement should be altered near the boiler. Others say that all risers should be taken off the side. Recently I met an engineer who claimed that perfect circulation could be had if the take-offs were varied for each set of risers depending on its distance from the boiler.

(4) The problem of getting an even distribution of heat throughout the house with but a slight temperature difference depends largely on questions 2 and 3, it being assumed, of course, that the radiators have been properly designed. A method should be found where the temperature would rise uniformly throughout the house when starting up the system.

(5) Recessing of radiators in new homes is usual practice now. What effect does this have on the radiator? Should the radiator be increased to allow for this condition or should the recess be designed so that it will offset the loss of heat due to the grille, etc.? Which method is the more advantageous?

(6) With the advent of the new fin type radiation the manufacturers have been advocating its use for hot water. Is it practical? What should the architect guard against in attempting to buy such units? What should the unit itself contain to overcome any tendencies for excessive accumulation of air?

(7) If fin type radiation is used should the mains and risers be re-designed to suit them? The manufacturers claim that a standard design of piping is sufficient to give satisfactory results. From actual experience I have discovered that this is not the case.

(8) Are orifices necessary as recommended? If the system were perfectly designed there should be no need for them and are they not an admission that the piping has been wrongly sized?

(9) Many engineers contend that the smaller the mains the faster the circulation. On the other hand, others contend the larger the mains the better the circulation. I have been advised that in Germany small size pipes are used and that they have developed pipe sizing to a high state of perfection.

(10) There appears to be no set temperature at which all engineers agree water should be heated and maintained. Temperature recommendations vary from 160 to 210 F. What temperature is the best and the most economical to operate a hot water system?

Many other questions could be asked but it seems to me that any book dealing with the subject of hot water heat should certainly offer positive solutions to the foregoing 10 points.

Answers should not be worded in such a manner as to make them hard to understand but should be as simple as possible, devoid of complex formulae and complicated theory that would require a highly trained specialist to understand. It is true that a book written for engineers need not necessarily take architects into account, but it is just as true that if it were so presented that architects might readily understand its contents, its value would be increased tremendously.

Hot water heating may some day be discarded. For the present it is with us. Few people know much or anything about it. Certainly no organization is better equipped to furnish authoritative data on the subject than the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS and it is with this aim in view and with a spirit of helpfulness that this paper has been written.

PROFESSOR SMITH: The first question raised by Mr. Bernhardt is that the tables are perhaps too complex. The heating and piping contractors have adopted a modification which is much simpler. I feel myself that they made their tables a little too compact; especially that they do not have enough orifice sizes to be sure that the circulation in the mains will not be upset. However, I think their tables will give good results in most cases and, of course, they can be modified from time to time as seems desirable.

The next question asks the best method of connecting radiators to mains or risers. The exact method followed does not make much difference if the system is properly balanced. Serious circulation troubles develop only when the pressure in a return main or a return riser gets above the pressure in the corresponding supply main or riser. Then a radiator starts backward and trouble begins. Friction a trifle too high in any radiator will do little harm; the temperature differential rises slightly, but the average temperature does not change much. It is perhaps best to take the supply off the top and the return off the side when convenient, but the important thing is to get the sizes of the pipes and the orifices correct.

Even distribution through the home is one of the things sought for in the tables. The radiators farthest from the heater have a temperature difference across them of about 22½ F and those nearest the heater a temperature difference of 37½ F, the idea being to keep the average radiator temperatures about the same and to compensate for the cooling in the mains. The tables are intended primarily for bare pipes.

The questions about recessing the radiators and the fin type radiators are outside the scope of the experiments. However, it appears to be just a matter of whether the resistance of such radiators varies a great deal from the resistance of the cast-iron radiators. Probably they can be used all right.

Mr. Bernhardt said that the orifices are an admission that the piping is not properly sized. The orifices are put in because there are not enough pipe sizes to make it possible to properly balance a system. It would be impractical to have so many pipe sizes and the orifices have to be used to secure the proper friction through each radiator circuit.

As to the question of whether one obtains faster or slower circulation with larger mains, there is a best size for the mains. If they are too large, there is too much cooling in the basement, and they are too expensive. It would be absolutely necessary to cover them. If the mains are too small, friction is too large and the pressure head generated by the heater is not sufficient to keep the pressure in the supply main above the pressure in the return main and circulation troubles are almost certain to develop. It is better to have the mains too large than too small.

The answer to the water temperature question raised by Mr. Bernhardt depends upon a number of variable factors. From the standpoint of the comfort of the room alone the lower the temperature of the water and the larger the radiators, the better will be the results because the temperature difference from floor to ceiling is certain to be less because a larger volume of air is heated to a lower temperature. Of course, the matter of the radiator being in the way and the cost of it will have to be considered also, so that there is no fixed standard. It also makes a difference where the system is to be used.

A. A. ADLER: Professor Smith's paper appears to have much merit in assisting designers in proportioning pipe sizes for hot water heating systems. However, cer-

tain cases may arise in which more refined computations are desirable to insure correct distribution.

While in general hot water heating systems have an inherent tendency to keep in balance despite slight departures from correct design, nevertheless attention should be directed to the influence of heat loss from the piping itself on the changes in the head which brings it about.

The effect of heat loss from piping is usually important in low buildings such as bungalows and 1-story factory buildings where the horizontal runs are long and the circulating head limited. Briefly the heat loss of the piping tends to increase the circulating head and, though it may seem strange to say, it makes little difference whether the bulk of the heat loss takes place on the flow or the return side of the circuit.

The subject is too broad to discuss here but adequate treatment is given by Rietschel-Brabbee in *Heating and Ventilation.* In unusual systems a computation should be made of the circulating head available, taking heat loss into consideration and balancing this against the resistance encountered in the circuit.

R. S. FRANKLIN: I have been much interested in the author's work. I have been on the Standards Committee of the *Heating and Piping Contractors National Association* for a number of years and this committee developed standards on which the certified heating program of that organization has been founded. It has therefore been necessary for this committee to put out sound information in a simple manner that could be readily applied. We have been urged for the last 10 years to make available information on hot water that would be safe and practical but the committee had been unable to do so until a year and a half ago when we first heard of Professor Smith's work in Philadelphia. The committee met at that time with Professor Smith and discussed his work. To test its application about a dozen different installations were made based on Professor Smith's data. All these installations operated successfully.

One particular installation I investigated had faulty circulation in a service wing. After obtaining some data from the existing job (no plans were available), I applied Professor Smith's data and made up a list of orifices, which I gave to a steam fitter with instructions to break the unions on supply valves to the specified radiators and install the orifices. As this cured the trouble, we became convinced that we could apply Professor Smith's data generally. As he has said, we have published a short-cut based on his data. Although boiled down to the point where it isn't theoretically sound, the data was used successfully in two recent installations. We believe that Professor Smith has made one of the most important of recent contributions to the practical application of hot water circulation.

PERRY WEST: For the benefit of the record and the benefit of our visitor from Philadelphia the answers to all of his questions will be found in THE GUIDE. What the architect needs is an engineer, or a heating and piping contractor, to give him the proper interpretation.

No. 903

USING GAS-VAPOR MIXTURES FOR HEATING PURPOSES

By C. A. DUNHAM,[1] CHICAGO, ILL.
MEMBER

THIS paper is the result of original research on the use of gas-vapor mixtures for heating purposes. The principle involved is that of burning gas in the presence of water and at pressures slightly below atmospheric, using a mixture of the combustion products and steam for conveying heat to the *radiators*.

DESCRIPTION OF APPARATUS

An especially designed generator with suitable gas controls was employed (see Figs. 1 and 2). The refractory with circular interior receives the gas flow introduced on one side near the bottom. The flame travel within this bowl is increased by means of the whirling action imparted by the rapid expansion of the burning gases injected into the refractory. The primary air is drawn into the burner at *a* by the natural inspirating action of the gas flow, and the secondary air is drawn in between the end of the burner and the refractory opening at *b* due to the lower pressure within the refractory. This air and gas mixture ignites off the end of the burner by means of a constant burning pilot and the flame passes through the hole into the refractory which soon becomes heated, and perfect combustion takes place before the flame reaches any cooler surfaces.

It will be observed in Fig. 1 that the generator has a double wall and that the dome of the inner wall is open at a point directly below the supply outlet of the generator. When the water level between the walls rises to an overflowing point (due to the expansion by heat) the water drips into a pan located directly below the opening, as indicated. This pan in turn overflows on to the flash plate which is slightly cupped to hold a shallow supply of water. Any surplus water passes to waste by dripping off the edge and down into the drain through the overflow pipe.

A complete system as it was applied to a house is shown diagrammatically in Fig. 3. A one-tenth horsepower electric motor operates an exhauster which

[1] President, C. A. Dunham Co., Chicago, Ill.

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in turn exerts a suction through the *radiators* and generator. It starts operating when the room push button is pressed and simultaneously brings on the gas for ignition by the constant burning pilot. This push button may be further supplemented to operate under the control of a room thermostat.

After the gas is ignited the sheet of water reaches the flash pan and is giving off vapor with no appreciable lapse of time. The products of combustion,

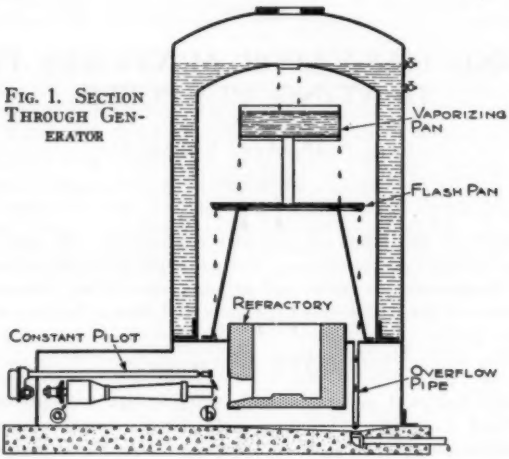

FIG. 1. SECTION THROUGH GENERATOR

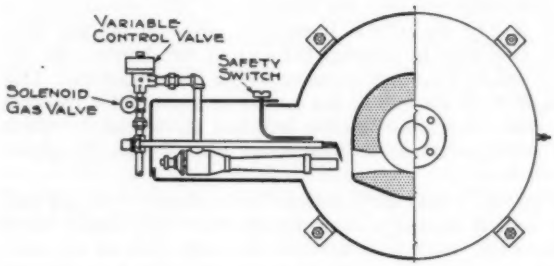

FIG. 2. TOP VIEW OF GENERATOR

after mixing with the vapor, are drawn immediately into the *radiators* where the vapor is condensed, the gases cooled, and the heat thus passed to the room being heated. The condenser vapor is returned to the water space between the walls of the generator.

The body of water which lies back of the inner wall of the generator and in direct contact with it, responds to the initial heat input by expanding, thus causing a trickling or overflowing into the vaporizing pan and then dripping

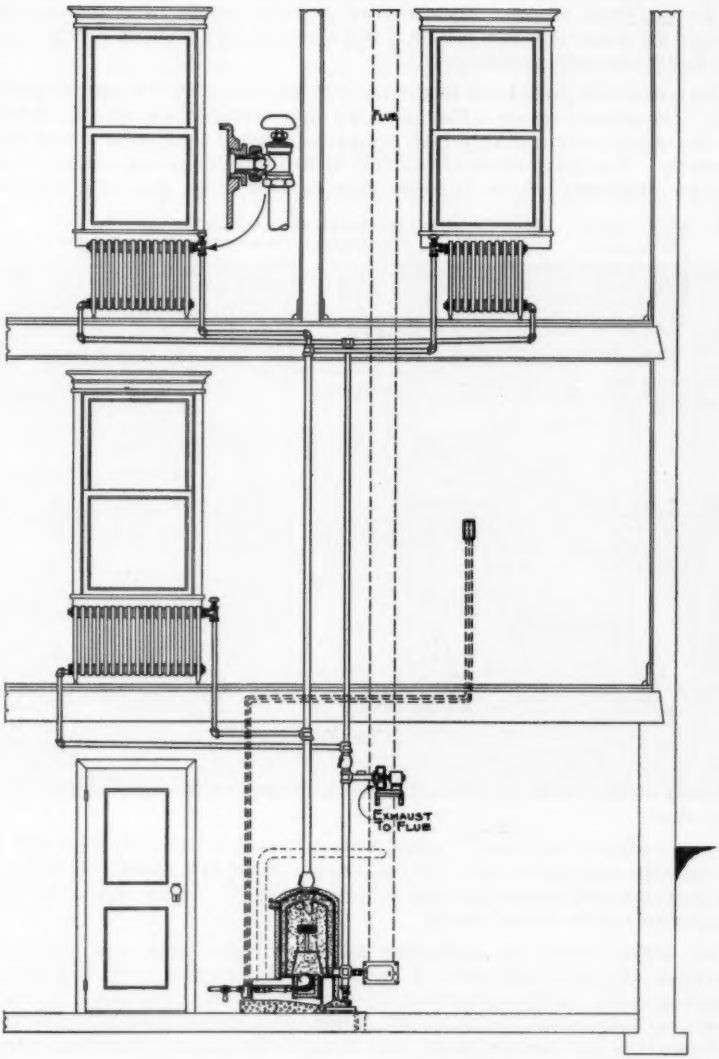

FIG. 3. COMPLETE SYSTEM APPLIED TO A HOUSE

to the flash pan below. This flow is made continuous by the constant return of the condensed vapors. These heating surfaces coupled with the surface area of the drops of water, present a large heating area and cause surprising results in vaporizing possibilities.

The question might arise as to why water is used instead of the gases to carry heat. The primary reason is that gas is not an economical heat vehicle. Without the water content a very high temperature would be created within the generator. The gases which are carried to the heat dissipating medium then bear no established relation in temperature to those in the generator, and the

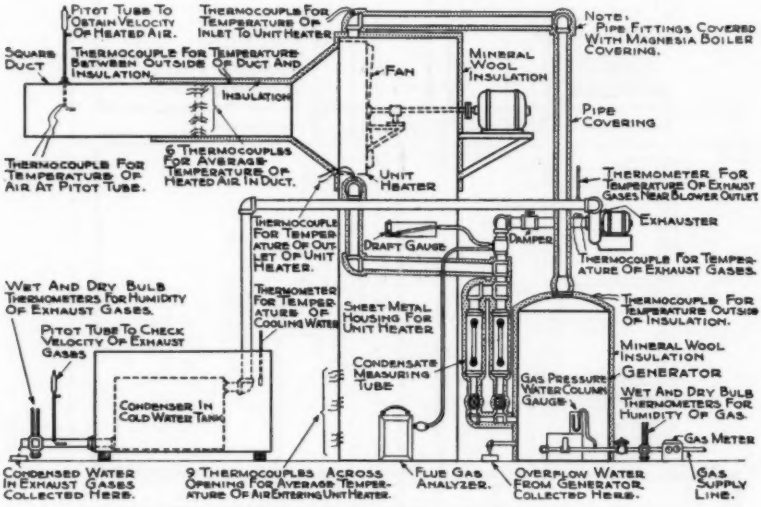
PITOT TUBE TO OBTAIN VELOCITY OF HEATED AIR.
THERMOCOUPLE FOR TEMPERATURE OF INLET TO UNIT HEATER
NOTE: PIPE FITTINGS COVERED WITH MAGNESIA BOILER COVERING.
SQUARE DUCT
THERMOCOUPLE FOR TEMPERATURE BETWEEN OUTSIDE OF DUCT AND INSULATION.
FAN
MINERAL WOOL INSULATION
INSULATION
PIPE COVERING
THERMOCOUPLE FOR TEMPERATURE OF AIR AT PITOT TUBE.
6 THERMOCOUPLES FOR AVERAGE TEMPERATURE OF HEATED AIR IN DUCT.
UNIT HEATER
THERMOMETER FOR TEMPERATURE OF EXHAUST GASES NEAR BLOWER OUTLET
EXHAUSTER
THERMOCOUPLE FOR TEMPERATURE OF OUTLET OF UNIT HEATER.
DRAFT GAUGE
DAMPER
THERMOCOUPLE FOR TEMPERATURE OF EXHAUST GASES.
WET AND DRY BULB THERMOMETERS FOR HUMIDITY OF EXHAUST GASES.
THERMOMETER FOR TEMPERATURES OF COOLING WATER
SHEET METAL HOUSING FOR UNIT HEATER
THERMOCOUPLE FOR TEMPERATURE OUTSIDE OF INSULATION.
PITOT TUBE TO CHECK VELOCITY OF EXHAUST GASES
MINERAL WOOL INSULATION
CONDENSATE MEASURING TUBE
GENERATOR
GAS PRESSURE WATER COLUMN GAUGE
WET AND DRY BULB THERMOMETERS FOR HUMIDITY OF GAS.
CONDENSER IN COLD WATER TANK
GAS METER
CONDENSED WATER IN EXHAUST GASES COLLECTED HERE.
9 THERMOCOUPLES ACROSS OPENING FOR AVERAGE TEMPERATURE OF AIR ENTERING UNIT HEATER.
FLUE GAS ANALYZER.
OVERFLOW WATER FROM GENERATOR COLLECTED HERE.
GAS SUPPLY LINE.

FIG. 4. COMPLETE TEST SET-UP

distance of the travel of gases at given temperatures through the piping is very short.

A test without any surplus water content showed a temperature of 600 F at the outlet of the generator. At the entrance to the first *radiator* 20 ft from the generator, the temperature was only 90 F, and the *radiator* would not heat completely without undue forcing.

By adding water, the conducting capacity of the pipes was materially increased and the temperature of the generator decreased to a level corresponding more closely to the heat input to the farthest *radiator*. In other words, an appreciable portion of the sensible heat in the gases was absorbed as latent heat and conveyed to the most distant points by the better heat carrying medium created by the additional water content.

In operation, practically all vapor returns to the generator after having given up its heat by condensing. The hydrogen content of the gas by combination with oxygen through combustion processes provides a surplus of water

beyond that required by the generator to form the heating medium and above the slight losses through the exhausted gases; this surplus water passes away through the drain. All noncondensable gases when cooled are discharged to the flue by the exhauster. A recent test was made to obtain the efficiency of the generator and also a heat balance to account for all the heat losses of such a type of heating system.

TEST PROCEDURE

Fig. 4 is a diagram of the complete test set-up. Figs 5 and 6 show end and side views of the apparatus. A unit heater was installed in the regular floor

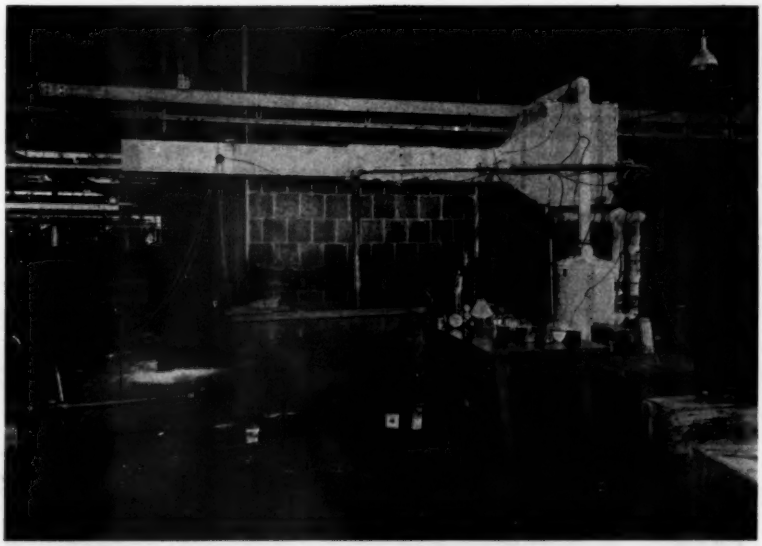

FIG. 5. SIDE VIEW OF TEST APPARATUS

stand with the outlet connected to a duct approximately one foot square. As shown in these figures the generator, unit heater and a portion of the duct were insulated.

Temperatures were taken by means of thermocouples in conjunction with the Leeds and Northrup type K potentiometer. Where extreme accuracy was not essential, temperatures were taken into calibrated glass stem thermometers. The flue gases were analyzed frequently throughout the test by means of the improved Hays apparatus.

In order to insure uniform temperatures the apparatus was operated continuously for a period of several hours before beginning the test. The readings were recorded for two more hours, an average of five readings, one each half hour, being used as the basis for the results of this report.

DATA AND CALCULATIONS [1]

(1) *Heat Value of Gas.* The heat value of the gas was obtained from an analysis made by the Northern Indiana Public Service Company, the net heat content averaging 577 Btu per cubic foot. The composition of the gas was as follows:

CO_2	=	1.8 per cent
O_2	=	2.8
N_2	=	16.0
CO	=	6.0
H_2	=	33.2
CH_4	=	35.1
C_2H_6	=	1.5
Illuminants	=	3.6
Total	=	100 per cent

[1] Symbols used in this paper follow:
A = internal area of pipe or duct, square feet
b = absolute pressure, inches of mercury
h = velocity head, inches of water
k = a constant
Q = flow or discharge, cubic feet per minute
t = temperature of air, degrees Fahrenheit
V = velocity, feet per minute
W = flow or discharge, pounds per hour

FIG. 6. END VIEW OF TEST APPARATUS

The gross heat value on this basis was 641.1 Btu per cubic foot.

(2) *Correction of gas as metered to standard conditions of 60 F, 30 in. mercury and 0 per cent humidity.*

Conditions of gas at meter during the test:

Average wet-bulb temperature	55.1 F
Average dry-bulb temperature	71.6 F
Pressure above atmosphere	4¾ in. water
Atmospheric pressure	29.50 in. mercury
Absolute pressure of the gas	29.85 in. mercury

Consider vapor pressure in the gas equal to vapor pressure in air having the same wet-bulb and dry-bulb temperatures = 0.3 in. mercury. Absolute pressure of dry gas as metered = 29.85 − 0.3 = 29.55 in. mercury. Correction factor for converting cubic feet of gas as metered to dry gas at 60 F and 30 in. mercury $= \frac{460+60}{460+71.6} \times \frac{29.55}{30} =$ 0.9635. Average gas consumption as metered during test = 124.4 cu ft per hour. Dry gas at standard conditions = 124.4 × 0.9635 = 119.9 cu ft per hour.

(3) *Water vapor supplied to the system during the test.*

(*a*) By gas supply

Condition of gas entering burner: Wet-bulb temperature, 55.1 F; dry-bulb temperature, 71.6 F.

Consider water content of gas equal to that of air at same wet- and dry-bulb temperatures = 0.0004 lb per cubic foot of gas as metered. Weight of water in gas supply = 124.4 × 0.0004 = 0.050 lb per hour.

(*b*) By air supply

Conditions of air used during test: 71.2 F; 29.50 in. mercury; 38 per cent relative humidity.

Saturated vapor pressure at 71.2 F = 0.7695 in. mercury. Density saturated vapor pressure at 71.2 F = 0.001201. Vapor pressure in air as used = 0.38 × 0.7695 = 0.292. Pressure of dry air = 29.50 − 0.29 = 29.21.

Volume of dry air at 60 F and 30 in. mercury used per hour = 808.2 cu ft. [See item (5).] Volume of air as used $= \frac{460+71.2}{460+60} \times \frac{30}{29.21} \times$ 808.2 = 847.9 cu ft per hour. Moisture content of air as used = 0.38 × 0.001201 = 0.0004564 lb per cubic foot. Weight of water in air supply = 847.9 × 0.0004564 = 0.387 lb water per hour.

Total water vapor supplied to system from moisture content in the air and by the gas supply = 0.050 + 0.387 = 0.437 per hour.

(4) *Water produced by combustion.* A vacuum of one inch of water existed at the top of the generator connection. The combustion of one cubic foot of the dry gas at 60 F and 30 in. mercury produces 1.185 cu ft of water at the same conditions. The theoretical volume if water vapor were a perfect gas at standard condition = 1.185 × 119.9 = 142.1 cu ft of H_2O produced per hour. Weight of water produced by combustion = 142.1 × 0.04758 = 6.76 lb per hour.

(5) *Per cent excess air.*

Average analysis of flue gas:

CO_2	=	8.7 per cent
O_2	=	3.7
CO	=	0
N_2	=	87.6
Total	=	100.0 per cent

Excess O_2 per cubic foot of flue gas = 0.037 cu ft. Necessary O_2 per cubic foot of flue gas = 0.226 − 0.037 = 0.189 cu ft.

Per cent excess O_2 and air = $\frac{0.037}{0.189}$ = 19.6 per cent. Air required for combustion of one cubic foot of gas plus excess air = 5.633 × 1.196 = 6.741 cu ft. Dry air supplied = 6.741 × 119.9 = 808.2 cu ft per hour, at standard conditions.

(6) *Non-condensable flue gases per hour.* Total volume of non-condensable flue gases per cubic foot of dry gas burned in test equals volume of gases produced by combustion plus excess air = 5.308 + 5.633 × 0.196 = 6.418 cu ft. Non-condensable flue gases per hour = 6.418 × 119.9 = 769.5 cu ft.

(7) *Water vapor which would have escaped per hour during test if gas had burned at an efficiency of 100 per cent.* If gas were burned in air at an efficiency of 100 per cent under standard conditions, perfect combustion would be obtained without excess air, and the products of combustion would be cooled to 60 F at 30 in. mercury pressure. In other words, the gross heat value of the gas would be obtained. The volume of non-condensable flue gases per cubic foot of dry gas burned at 100 per cent efficiency = 5.308 cu ft. (Gas volume at standard condition.) Gases remain saturated at 60 F. Saturated vapor pressure at 60 F = 0.5214 in. mercury. Pressure of dry gases = 30.0 − 0.52 = 29.48 in. mercury. Volume of saturated gases = $\frac{30}{29.48}$ × 5.308 = 5.402 cu ft. Density of saturated vapor at 60 F = 0.000829 lb per cubic foot. Water vapor escaping per cubic foot of gas burned = 5.402 × 0.000829 = 0.004479 lb. Water vapor which would have escaped during test if gas had burned at an efficiency of 100 per cent = 119.9 × 0.004479 = 0.54 lb per hour.

(8) *Water carried to atmosphere by flue gases.* Non-condensable flue gases per hour at standard conditions [from item (6)] = 769.5 cu ft. Gases enter the atmosphere saturated at 93.6 F. Atmospheric pressure = 29.50 in. Saturated vapor pressure at 93.6 F = 1.589 in. Pressure of dry gases = 29.50 − 1.59 = 27.91 in. mercury. Volume of gases entering atmosphere = $\frac{460 + 93.6}{460 + 60} \times \frac{30}{27.91} \times 769.5$ = 880.6 cu ft per hour. Vapor content = 0.002375 lb per cubic foot. Water vapor carried to atmosphere = 880.6 × 0.002375 = 2.09 lb per hour.

(9) *Heat loss of system through liquid and vapor loss.* This is equal to the water whose heat of liquid above 72 F and heat of vaporization is lost.

Water collected at condenser outlet plus water escaping to gas minus water vapor which would have escaped through flue atmosphere minus water vapor supplied to system by air and if gas had burned at 100 per cent efficiency = 4.40 + 2.09 − 0.05 − 0.387 − 0.54 = 5.51 lb per hour. This entered the exhauster as vapor at a temperature of 130 F. Total heat of vaporization lost = 5.51 × 1,020 = 5,620 Btu per hour. Heat of liquid lost = 5.51 × (130 − 72) = 320 Btu per hour. Heat of liquid of overflow from generator is lost = 0.47 (150 − 72) = 36 Btu per hour. Total heat lost as heat of vaporization and heat of liquid = 5,620 + 320 + 36 = 5,976 Btu per hour.

(10) *Heat lost from system as sensible heat of dry gases.* Heat lost from system as sensible heat of dry gases based on flue gas analysis = 880.6 × 0.06850 × 0.24 × (130 − 72) = 839.7 Btu per hour.

(11) *Total accountable heat loss.*

Latent heat	= 5,620 Btu per hour [Item (9)]
Heat of liquid	= 320 Btu per hour [Item (9)]
Sensible heat of gases	= 840 Btu per hour [Item (10)]
Heat loss through flue	= 6,780 Btu per hour
Total accountable heat loss	= 6,780 + 36 Btu at overflow

Total accountable heat loss = 6,780 + 36 Btu at overflow [Item (9)] = 6,816 Btu per hour.

(12) *Heat produced by combustion of flue gas.* Gross heat value of fuel gas burned per hour = 119.9 × 641.1 = 76,841.6 Btu [See items (1) and (2).]

(13) *Velocity of flue gas and vapor through 2-in. pipe discharge to atmosphere.* Condition of gases at point velocity head was measured: saturated at $t = 93.6$ F and $b = 29.50$ in.

$V = k\sqrt{h}$

$k = 952.8 \times \sqrt{\frac{460 + 93.6}{29.50}} = 4{,}125.6$

Average $\sqrt{h} = 0.212$ (h measured at center of pipe)

Correction for $\sqrt{h}$ taken as 0.933

Correction for density $= \sqrt{\frac{0.07068}{0.07088}} = \sqrt{0.9972} = 0.9986$

$V = 4{,}125.6 \times 0.175 \times 0.933 \times 0.9986 = 676.5$ ft per minute.

(14) *Rate of flue gas discharge to atmosphere* $= Q = A \times V$.
$A = 0.0233$ sq ft (for 2-in. pipe). $V = 676.5$ ft per minute [from item (13)]
$Q = 0.0233 \times 676.5 = 15.76$ cu ft per minute or 945.6 cu ft per hour.

(15) *Density of flue gases.* Density of dry flue gases at 93.6 F and 29.50 in. = 1.0244 × 0.07068 = 0.07240 lb per cubic foot. Pressure of saturated vapor at 93.6 F = 1.589 in. of mercury. Density of saturated vapor at 93.6 F = 0.002375 lb per cubic foot. Pressure of dry gases in mixture = 29.50 − 1.59 = 27.91 in. of mercury. Density of dry gases at 93.6 F and 27.91 in. mercury $= \frac{27.91}{29.50} \times 0.07240 = 0.06850$ lb per cubic foot. Weight of one cubic foot of the gas vapor mixture = 0.06850 + 0.00238 = 0.07088 lb.

(16) *Water carried to atmosphere by flue gases.* Gases and vapor discharged at 93.6 F and 29.50 in. mercury = 945.6 cu ft per hour. The water vapor per cubic foot of mixture = 0.002375 [item (15)]. Water carried to atmosphere by flue gases = 945.6 × 0.002375 = 2.25 lb per hour.

(17) *Total heat of vaporization lost.* By procedure similar to that given in [item (9)], the total heat of vaporization lost = 5783 Btu per hour, and the heat of liquid = 329 Btu per hour lost in the flue gases.

(18) *Heat lost as sensible heat of non-condensable flue gases* = 945.6 × 0.06850 × 0.24 × (130 − 72) = 901.7 Btu per hour.

(19) *Heat loss through flue and from overflow.*

Latent heat	= 5783 Btu per hour
Heat of liquid	= 329
Sensible heat of gases	= 902
Total loss through flue	= 7014 Btu per hour
Loss from overflow	= 36
Flue and overflow losses	= 7050 Btu per hour

(20) *Velocity of air in discharge duct* $= V = k\sqrt{h}$

For dry air $k = 952.8\sqrt{\frac{460 + t}{b}}$

Condition of air in duct during test at point velocity head was measured: Average $t = 97.6$ F; $b = 29.50$ in.; relative humidity = 12½ per cent.

$$k = 952.8\sqrt{\frac{460 + 97.6}{29.5}} = 4134.6$$

Average $\sqrt{h} = 0.648$ (h taken at center of duct). Correction factor for $\sqrt{h} = 0.933$. Density of dry air at 97.6 F and 29.50 in. mercury = 0.070186 lb per cubic foot. Density of air at 12½ per cent relative humidity, 97.6 F and 29.50 in. mercury = 0.069986 lb per cubic foot.

Correction factor for humidity of test air $= \sqrt{\frac{0.070186}{0.069986}} = 1.0014.$

$V = 4{,}134.6 \times 0.648 \times 0.933 \times 1.001 = 2{,}502.0$ ft per minute.

(21) *Rate of air flow through discharge duct* $= Q = A \times V$. A (of duct) = 0.989 sq. ft. V [from item (20)] = 2502.0 ft per minute. $Q = 0.989 \times 2502 = 2474.4$ cu ft per minute, or 148,464 cu ft per hour. Density of air under test conditions (12 per cent relative humidity, 97.6 F, 29.5 in. mercury) = 0.069986 lb per cubic foot. $W = 148{,}464 \times 0.069986 = 10{,}390$ lb per hour.

(22) *Heat delivered by radiator* equals pounds of air per hour times average specific heat times temperature rise in degrees Fahrenheit. Average specific heat of test air = 0.243. By test it was found that the air was heated 1.5 deg by the action of the fan. Thus the actual rise of temperature due to generator = 101.1 − (72.7 + 1.5) = 26.9 deg. Heat delivered by radiator = 10,390 × 0.2430 × 26.9 = 67,920 Btu per hour.

(23) *Heat balance*

Heat delivered by *radiator*		67,920 Btu per hour
Lost through the flue		
(*a*) by flue gas analysis	6780	
(*b*) by Pitot tube measurements	7014	
Average		6,897
Lost in overflow		36
Unmeasured losses, including heat loss through insulation		1,989
Total heat output of unit		76,842 Btu per hour
Total heat input to unit		76,842 Btu per hour

(24) *Calculation of total heat loss through flue if flue gases had been cooled to 97 F at the exhauster:* Non-condensable flue gases per hour (volume at standard conditions) = 769.5 cu ft [item (6)]. Pressure of gases at inlet to exhauster = 3.5 in. water below atmospheric or 29.50 − 0.26 = 29.24 in. mercury, absolute. Saturated

vapor pressure at 97 F = 1.763 in. mercury. Pressure of non-condensable gases = 29.24 − 1.76 = 27.48 in. mercury. Volume of saturated gases entering exhauster = $\frac{460 + 97}{460 + 60} \times \frac{30}{27.48} \times 769.5 = 899.8$ cu ft per hour. Density of saturated vapor at 97 F = 0.002621 lb per cubic foot. Water vapor entering flue = 899.8 × 0.002621 = 2.36 lb per hour.

Water vapor whose heat of vaporization and heat of liquid to 72 F is lost = total vapor entering exhauster minus water vapor supplied to system by air and gas minus water vapor which would have escaped if gas had been burned at 100 per cent efficiency = 2.36 − 0.05 − 0.387 − 0.54 = 1.38 lb per hour. Total heat lost as heat of vaporization: 1.38 × 1038.2 = 1432.7 Btu per hour. Heat lost as heat of liquid = 1.38 (97 − 72) = 34.5 Btu per hour. Heat lost as sensible heat of non-condensable gases = 880.6 × 0.06850 × 0.24 × (97 − 72) = 361.8 Btu per hour. Total heat lost through flue = 1432.7 + 34.5 + 361.8 = 1829 Btu per hour. Heat loss in overflow from generator = 36 Btu per hour [item (8)]. Total heat loss through flue if gases had been cooled to 97 F at the exhauster = 1865 Btu per hour.

RESULTS

With an exhaust temperature of 130 F at exhauster outlet the efficiency of the system = $\frac{\text{heat supplied} - \text{heat lost}}{\text{heat supplied}} = \frac{76{,}842 - 6{,}816}{76{,}842} = 91.1$ per cent based on flue gas analysis.

For checking purposes, the efficiency of the system based on Pitot tube readings and calculations is next determined as follows: $\frac{76{,}842 - 7{,}050}{76{,}842} = 90.8$ per cent. [See items (13) to (19) inclusive].

The efficiency of system based on duct method is next found from items (20), (21), (22) and (23). By the duct method the total heat accounted for leaving the system = 70,435 + 7025 + 36 = 77,496 Btu per hour. Efficiency of system = $\frac{\text{heat delivered by radiator}}{\text{heat supplied to system}} = \frac{67{,}920}{76{,}842} = 88.4$ per cent.

Based on outlet temperature of 97 F at the exhauster, which temperature was attained on actual installations, the efficiency of system based on flue-gas-analysis method [see item (24)] indicates a possible efficiency of 97.6 per cent.

DURABILITY

Past experience with the use of products of combustion associated with water vapors very probably causes one to ask the question, "What about corrosion?" Observations extending over a period of two years would indicate that where the fuel is natural gas, coal or coke oven gas, burned under the conditions of this test, the rate of penetration indicates a long life which is comparable to other types of heating systems. Examination of a generator interior, after almost a year of use with Chicago artificial gas, shows a degree of corrosion that is surprisingly small.

Practically all heating systems rust somewhat inside and usually at a definite rate in inches of penetration per year, depending of course upon the kind of metals and how much the system is used. Because some smokestack breechings have been known to rust in spots under certain circumstances after a comparatively short period of use, one naturally envisions the possible effect of circulating stack gases through pipes.

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All stack breechings do not give corrosion trouble nor do all fuels have the same percentage of harmful ingredients in them. Coal as a rule has a very high percentage of sulphur in its products of combustion, which, if combined with moisture and soot, causes mild acids to form and which ordinarily stimulate the rate of rusting above that caused by moisture alone. On the other hand, natural gas as marketed today contains little or no sulphur, and if properly burned produces no soot.

The relative effect of hydrogen-ion concentration on the corrosion of steel and with water which does not contain salts or impurities (which form protective coatings) indicates that at low temperatures the evolution of hydrogen does not increase until a *pH* value of approximately 4 obtains. The *pH* value of the water from various generators which have operated for considerable periods of time with different manufactured gases, showed a *pH* value greater than 4 when measured by the quin-hydrone electrode. The *pH* values obtained from the test set-up were well above 6 on the *pH* scale.

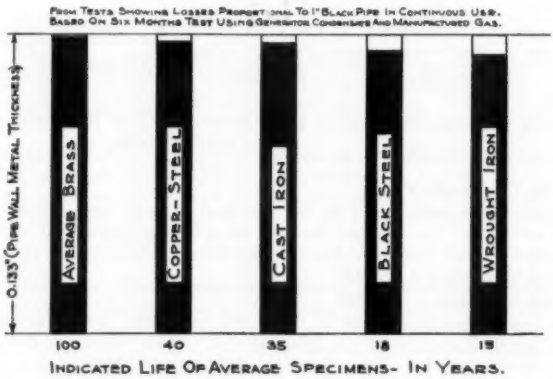

FIG. 7. PENETRATION OF STANDARD PIPE PER YEAR

In districts where a poor quality of manufactured gas is produced, it has been found that the hydrogen-ion concentration of the water under the conditions of operation with this generator did not increase enough to increase the rate of action over the neutral water range, and in no case did the water become *bad* from a classifying standpoint due to the acid or scale forming content.

The favorable showing on this corrosion question may be due to several factors, *i.e.*, the constant washing effect from the large volume of water and vapor, which prevents higher localized concentrations forming, and the continuous flushing through overflow. Also, the fact that the generator operates at a slightly sub-atmospheric pressure with reduced oxygen content (below that of ordinary air) with an increase of inert nitrogen, coupled with the fact that the volume of vapor fills the entire space not occupied by gas, may have a favorable bearing. Again with the perfect combustion which is possible under the controlled air and gas supply, no soot results, and sulphur compounds when present are notably less active at temperatures below 500 F.

The total amount of sulphuric acid which may be formed from such low sulphur fuels per season under these combustion conditions is found to be small and appears to be sufficiently diluted by the excess water produced to prevent excessive rusting.

As a matter of fact, natural gas is being piped into cities far remote from the source, and better means of purifying manufactured gas is rapidly coming into use, which, with better materials obtainable at low cost, seems to invite serious thought to this more direct method of heating. Tests of metal specimens in solutions of the concentrated generator water for a period of six months are shown in Fig. 7. It will be observed that the steel alloy pipe containing 0.25 per cent copper has more than double the life of ordinary wrought iron and steel under these conditions. This is believed to be due to a dense adherent type of rust which prevents scaling and slows up further attack.

There has been no evidence of any appreciable building up of harmful substances over a period of time. Neither has any soot been found or any scaling of pipes where copper steel alloy is used.

SUMMARY AND CONCLUSIONS

This paper is based on tests made on a specially constructed generator in which gas is burned in the presence of water, thus forming a gas-vapor mixture for heating *radiators*. The most important considerations may be summed up briefly as follows:

1. The elimination of loss in heat transfer through metals at point of heat generation.
2. The reduction in loss of heat from flue gases.
3. Absorption of sensible heat in the gas by conversion to latent heat in vapor which in turn makes it possible to convey heat long distances without appreciable loss in sensible temperature.
4. Reduced temperatures and pressures of gas-vapor, thereby minimizing corrosion as indicated by definite tests showing the rate of penetration per year to be within limits which admit of practical use.
5. Simplicity. An overall efficiency for gas fuel in the 90 per cent range.

It is not the desire to convey the impression that all angles of this method of heating have been investigated. That is conceded as too large an undertaking to complete in two years' time. The first investigations were centered on *results* where it was apparent that additional time was required to prove the *causes*.

DISCUSSION

PERRY WEST: Will Mr. Dunham tell us whether the life of the pipe was based on the thickness back of the thread, as shown on the sketch, or was it based on the life at the base of the thread? There is where the pipe first gives out.

C. A. DUNHAM: The life was based on the thickness just back of the thread. At the base of the thread there would be perhaps 25 per cent less life. Threads made up tight will stand a great deal of corrosion before there is any leakage because the leakage is in and not out; the pressure is in and not out with this system.

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F. D. MENSING: Mr. Dunham, how many samples of pipe did you take and from how many sources were they obtained?

MR. DUNHAM: The samples that are shown here on the sheet were with Chicago gas taken from the outlet of the generator. We have, however, analyzed the gas in southern California, Iowa, Indiana, Chicago and New York, and we find that there is quite a difference in gases. There is a difference in the corrosion effect, but we did not find any appreciable difference due to the location in which these sample test pieces were placed. We submerged them in the generator water; we put them in the flow of gases of the supply and in the return, but we found little difference in the corrosive action.

No. 904

ESSENTIAL ELEMENTS FOR DETERMINING HEATING PLANT REQUIREMENTS

By F. B. Rowley,[1] Minneapolis, Minn.
MEMBER

OFTEN, that which seems to be a comparatively simple problem becomes involved and controversial because of failure to consider all of its related parts. Thus, the problem of determining the requirements for heating plants appears to be without any specific challenge. It is only a matter of supplying enough heat to maintain a comfortable living temperature and to take care of any heat losses from the building. Practically every one in the heating profession has a definite conception of the requirements of this problem as a whole; yet, when the integral parts are considered, there are wide differences of opinion. The purpose of this paper is to analyze the problem and to consider some of the causes for these differences.

Without quoting specific reference, free use has been made of the facts obtained by researches at such places as the A. S. H. V. E. Research Laboratory, the U. S. Bureau of Mines, the University of Illinois, the University of Wisconsin, and other institutions and research laboratories. An effort has been made to correlate the various parts of the problem and to arrive at a logical solution which would omit the unknown factors so often applied to heating plant design.

Insofar as the direct heat supply from the plant is concerned, there are at least four factors which have a direct, or, at least, an indirect bearing on the question. These are:

1. The air temperatures required within the space to be heated.
2. The heat utilization or the efficiency of the heat-distributing system used.
3. The unit of measure by which heaters are rated.
4. The method used for calculating heat losses from the building.

Other angles of the problem may present themselves, but it appears that these four questions are the bases for the greatest differences of opinion.

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Presented at the Semi-annual Meeting of the American Society of Heating and Ventilating Engineers, Swampscott, Mass., June, 1931.

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AIR TEMPERATURES REQUIRED

Everyone wants to be comfortable, and lack of comfort is probably the major cause for criticism of heating plants. The most direct relation which the heating plant bears to comfort is that of providing proper conditions so that the body can regulate its internal temperatures. The body is continually generating heat, and this heat must be dissipated at such a rate that the temperature of the body will be maintained within very narrow limits. This heat may be given directly to the surrounding air by convection and conduction, it may be absorbed by the evaporization of moisture, or it may be given off by radiation.

The relation of air conditions to the amount of heat given off directly to the air has been very carefully worked out by the A. S. H. V. E. Research Laboratory in cooperation with the U. S. Bureau of Mines. The relation of the amount of heat given off by radiation to the temperatures of the surrounding objects has also been given consideration. It is evident that if the surrounding conditions are such that the body radiates a large proportion of its heat, the amount given off directly to the air must be reduced, and, likewise, if the radiation factor is reduced, the amount given off by convection must be increased.

In practice, air having a temperature of 70 F has been considered as the average optimum condition to be aimed at by the heating plant designer. However, it can readily be seen that if the wall temperatures are very cold, such as might be the case if a person were surrounded largely by single glass windows on a sub-zero day, then it would be necessary that the air temperatures be materially higher in order to counterbalance the excess amount of heat or proportion of heat given off by direct radiation. On the other hand, if a person were surrounded entirely by inside walls which might be at a very high temperature, the air for comfort would then have to be a considerable amount below 70 F to satisfy the heat-dissipating requirements of the body.

While it is evident that a very wide range of temperatures may be required for comfort under different surrounding conditions and thus the comfort requirements may throw a variable factor on the heating plant, it is also evident that some practical, average condition must be assumed for use in the design of heating plants. This practical average has been almost universally accepted as 70 F.

Experience has shown that, with many types of heaters, there is a variation in room temperatures from floor to ceiling of as much as 15 deg to 25 deg. It will thus make a material difference as to how high from the floor the 70-F temperature is to be measured. Until recently, the most common point of measurement has been at 5 ft above the floor and not less than 3 ft from the outside wall, this being considered as the average breathing line. Recent investigators have considered this point too high, and 30 in. from the floor now seems to be an acceptable standard. If all radiators heated the room uniformly from top to bottom, the height from the floor at which temperatures were measured would make no difference either as to comfort or in the design of the heating plant. Unfortunately, however, many radiators do not have this uniform heating effect and, therefore, the load on the plant will be somewhat varied, depending upon the height assumed.

The effect of wall surface temperatures upon the radiant heat given off by the body should, undoubtedly, have further study in order to determine their effect on the comfortable living temperature, and, also, upon the requirements of the heating plant. The character of the wall surfaces also has something to do with the amount of heat given off by direct radiation from the body and, therefore, will affect the comfortable air temperatures required.

HEAT UTILIZATION

The efficiency with which heat is applied to the space to be heated has a direct bearing upon the heating-plant load. The requirements are to keep the body under comfortable living conditions as to temperatures of the air and of the surrounding objects. Since this condition is required at a distance of 30 in. from the floor, it is evident that excessively high temperatures at other parts of the room may result in a direct waste of heat, and, therefore, an overload on the heating plant. As has been previously pointed out, certain types of radiators will give temperatures from 15 deg to 25 deg higher at the ceiling than at the floor line. These high ceiling temperatures are useless, insofar as the comfortable conditions of the room are concerned, and for efficiency might well be avoided.

Take as an illustration a room with a 10-ft ceiling and with a 20-deg difference in temperature from floor to ceiling. With a straight line temperature variation and with 70 F at a point 30 in. from the floor, the floor line temperature will be 65 F and the ceiling temperature will be 85 F, or an average air temperature of 75 F which, with an outside temperature of zero would mean over 7 per cent additional heat loss through the outside walls. If the ceiling were exposed, the heat loss would be increased through the ceiling by slightly over 20 per cent as compared with the losses with ceiling air at a temperature of 70 F. The variations might be other than 20 deg, and the conditions surrounding the building, which would affect the loss by direct transmission and by infiltration, might be such as to increase the load on the plant anywhere from 0 up to 20 or more per cent, due to the inefficient application of the heat to the room.

This inefficiency in heat application is due mostly to the difference in temperatures from floor to ceiling and is caused largely by the type and location of the radiator or heating unit in the room. Any radiator which is so designed and so located that it will deliver air at warm temperatures to the top of the room will cause trouble, due to the stratification of the air, unless artificial means are used to break up this stratification and to cause the air to circulate. In cast-iron *radiation*, a low radiator of a given width has been found to give more uniform temperature distribution from floor to ceiling than a higher design of the same radiator. This is evidently due to the fact that the higher radiator delivers the air from the top at a higher velocity and a higher temperature. This air, in turn, reaches the ceiling at a temperature considerably above the average room temperature, remaining there to cause the uneven distribution. With the low radiator, the air is not thrown off with such high velocities or at such high temperatures, which gives it more of an opportunity to diffuse with the air in the room, and when it does reach the ceiling, insures that it will be at a lower temperature. Take the case of a wall type radiator placed under the window; experiments have shown that there is a very uniform

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temperature distribution from floor to ceiling. In this case, the warm air currents from the radiator pass up, meet, and diffuse with the cold air currents coming down from the window surface. These mixed air currents are at a much lower temperature than would be the case if the radiator were not placed under the window.

As far as the efficient distribution of convected air currents is concerned, the first step appears to be to prevent high temperature air from reaching the ceiling of the room. After it has once reached the ceiling, it cannot be brought down to the floor again by gravity current.

Besides the inefficiencies due to uneven temperature distribution, there may also be radiant heat losses. A part of the heat of most heating units is given off by direct radiation. This may vary from something over 30 per cent in ordinary cast-iron *radiation* down to very small percentages in some of the convector types of heating units. The effectiveness of radiant heat depends upon what happens to it after it leaves the radiator. Radiant heat does not affect the air directly but must strike some object which will absorb it and transmit it to the air by conduction and convection. If radiant heat strikes inside wall surfaces or objects, it should, in general, be effective in supplying

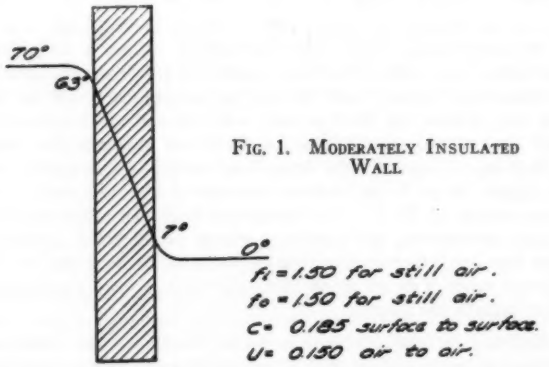

FIG. 1. MODERATELY INSULATED WALL

the heat requirements for the building. If, on the other hand, radiant heat strikes outside wall surfaces which are very cold, such as glass, its effectiveness may be greatly reduced.

As an illustration of what may happen to radiant heat when coming in contact with outside wall surfaces, refer to Figs 1 and 2. Fig. 1 represents a reasonably well insulated wall, the overall heat transmission coefficient (U) of which is 0.15, and Fig. 2 represents a single glass window with an overall heat transmission coefficient of 0.75 for still air. If, for Fig. 1, the inside air temperature is assumed to be 70 F and the outside, zero, with reasonably still air on both sides of the wall, the surface coefficients may be taken as 1.5 and the temperatures of the wall surfaces would be 63 F and 7 F, respectively. If radiant heat were to strike this wall on the inner surface, the effect would be to build up the inner surface temperature and increase the heat flow through the wall to that extent. As the wall temperature approached 70 F, the amount

of heat given to the wall by convection from the air would be decreased. With an extreme condition, in which most of the heat were given off by radiation, the wall temperature might reach 70 F. In this case, the heat loss through the wall would be increased by 11 per cent. This would be a maximum condition and would represent a radiator in which practically all of the heat was given off by direct radiation and all radiator surfaces exposed to outside wall surfaces, a condition which would be very unlikely. It is thus evident that radiant heat is not a serious factor when we consider walls which are well insulated.

Next, consider the wall represented by Fig. 2, in which case the glass has a very low internal heat resistance, the greater part of the resistance to heat flow being on the two surfaces. With air at 70 F and zero, respectively, on the two sides of the glass, it is reasonable to assume that the average temperature of the glass surfaces will be 35 F. In this case, if radiant heat were

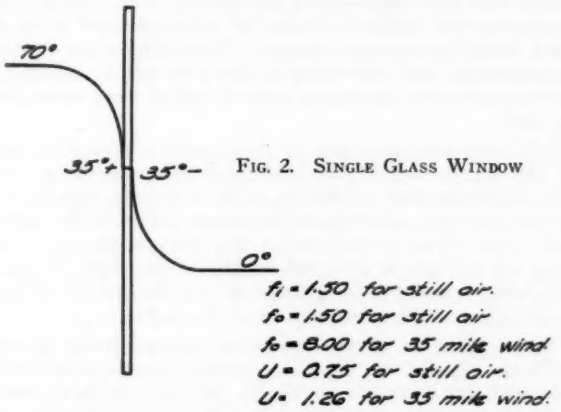

FIG. 2. SINGLE GLASS WINDOW

to strike and be absorbed by the inner surface, the temperature would rise and the heat loss through the glass would be increased substantially proportional to this rise in temperature. If an extreme case is assumed in which the inner surface is brought up to 70 F by radiant heat, the heat loss through the glass would be nearly doubled.

A further condition which might occur is that of a wind velocity on the outside of the surface, such as to increase the outer surface coefficient from 1.5 to 5.0 or more. If, under this condition, the inside surface temperature were raised to 70 F, the total heat loss through the thin wall might be increased by six or more times the original amount.

The exact amount of radiant heat which might be lost through thin walls depends upon several factors, such as surface coefficients, wind velocities, internal heat resistance of wall, and amount of radiant surface exposed to outside walls. In order that radiant heat may be effective, it should be received by objects in the room or by inside walls and not by outside walls which have very low thermal resistance. Moreover, it should be received by objects which

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are at a low elevation in the room in order to maintain the temperature distribution as low in the room as possible.

From the foregoing, it is evident that heat utilization may be an important factor in considering the load on the heating plant, but it is further apparent that the effectiveness of both convected and radiant heat is somewhat under the control of the designer. With the proper consideration for the laws governing heat distribution from heating units, a very high heat utilization factor should be obtained.

UNIT OF MEASURE FOR HEATERS

The question of heat utilization brings up that of a proper method to measure heating units. It is evident that the effectiveness of various types of heaters is different, due to the fact that they distribute their heat in the room by different methods. If this difference in effectiveness is not taken into account in rating or measuring the output of the unit, it will place an uncertain factor on the final load required for the heating plant. It is true that this difference may lie well within the factor of safety assumed in the design; yet it exists and should not escape attention. The question has received a great deal of consideration and discussion in the past and is especially pertinent when it comes to the selection of test methods and of a test room for the rating of heating units.

In general, there are two types of rooms which have been used for this purpose. The first is one in which the wall temperatures are substantially the same as the temperature of the air in the test room, and the heat dissipation from the room takes place by an interchange of air to the space surrounding the test room. The second type of test room is one in which the heat given off by the radiator is absorbed directly by the walls of the test room, these walls being cooled by refrigerated air on the outside or by means of spraying cold water on the outer surfaces of the walls.

In the first type of room, the convection currents from the radiator are somewhat disturbed, and the heat distribution from top to bottom of the room is changed, due to the air circulation into and out of the test room. If the test room is so arranged that there is a free circulation of air out of the top and in at the bottom, it is quite possible that a radiator which in practice would result in a large temperature difference from floor to ceiling would show only a nominal difference in the test room, due to the fact that the hot air is freely discharged at the top and thus not allowed to build up in temperature. These convection currents may not be such as seriously to affect the heat output from the radiator, but they may affect very materially the temperature distribution as measured throughout the different elevations of the room.

A second point which should be given consideration in this type of room is that of the absorption of radiant energy by the walls. Since all surrounding walls are substantially at the same temperature as the air in the room, they would virtually be inside walls and the radiant energy might be more effective than would be the case if the radiator were used in an actual room with wall surfaces below the temperatures of the surrounding air. In the majority of cases, the magnitude of this error is probably small, due to the fact that, in most radiators, only a small proportion of the heat is given off by direct radia-

tion, and of this, only a small proportion strikes the outside cold surfaces directly from the radiator. Furthermore, as has been shown in the discussion of Fig. 1, if these cold surfaces have any appreciable amount of internal resistance, the effect of the radiant heat losses is only a small proportion of the total radiant heat.

Taking everything into consideration, it is evident that this type of test room disturbs the temperature distribution factor very materially, but, in most cases, does not have any great effect upon the total heat output from the radiator. The room is not well adapted to test a radiator for heat distribution, but is a convenient form of test room for determining the total heat output by the condensation method. It is flexible and can be adjusted for different capacities.

Considering the second type of test room, this can be made to approach a practical condition, providing the walls are cooled in such a manner that their inner surface temperatures will equal those met in practice. For radiators giving off the greater portion of their heat by convection, it would appear to make no difference as to the internal heat resistance of the walls or as to the methods used in cooling them to bring the internal or inside surface temperatures to the proper degree. As long as these temperatures are properly maintained, the heat conducted from the air in the room to the surface of the wall by convection currents should be the same as met with under practical conditions.

If radiant heat is considered, the method of cooling the walls, and the amount of internal resistance in the test room walls may have a decided effect upon the efficiency or heating effect of the radiator. Take, for instance, the very thin metal wall which is cooled by circulating water over the outside surfaces to maintain an inside surface temperature at 63 F. In this case, there is an unlimited supply of 63 F water, which will maintain the internal surface temperature regardless of the amount of heat given to the wall. Thus, if radiant heat strikes this wall, it is absorbed by it without raising the inside temperature, and, therefore, does not retard the heat given to the wall by convection, which means that all radiant heat is virtually lost, without in any way affecting the air temperature within the room. If, in this type of test room, a radiator which gives off 30 per cent of its heat by radiation were tested and so placed that half of the radiant heat reached the cold walls, it would mean a 15 per cent loss without any heating effect to the air in the room. If this same radiant heat struck a test wall in which there was a large internal heat-resisting factor, it would serve to raise the inside surface temperature, and, therefore, would not be a direct loss. The direct loss would be reduced as the internal resistance of the wall was increased. If a convection type of heater were to be tested in this room, the only function of the test room cold walls would be to absorb the heat from the air, as the air temperature would be raised within the radiator itself and would not depend upon the radiant heat being intercepted by the wall surfaces and returned to the air in the room. It is conceivable that a radiating unit might be so designed as to emit practically all of its heat by direct radiation, in which case the total output from the radiator might be entirely ineffective if the radiant heat should strike one of the thin metallic walls cooled to a pre-determined inside surface temperature by an unlimited amount of circulating water on the outside surface. This radiant heat could not be so lost if the walls contained a practical amount of internal heat resist-

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ance. In this case, the surface temperatures of the wall would be increased until the heat was transmitted to the air in the room.

From the foregoing, it is evident that the type of test room is very important if radiators of different design and characteristics are to be tested for comparison. If heat distribution is to be one of the factors obtained from the test, the method of dissipating the heat by an interchange of air to the exterior of the room may lead to appreciable errors in the heating effect as indicated from the test. If the total heat supplied by the radiator is the only factor regardless of its effectiveness in heating the room, then the first type of testing room apparently has advantages and the heat output may be measured by the condensation method.

If heat distribution, as well as heat supply, is to be measured, the second type of room or the one in which the wall temperatures are maintained at some practical degree below those of the air for at least a part of the walls seems to have the advantage. If the units under test are designed to emit the greater part of their heat by convection currents, there should be no material difference as to the method of cooling the walls or as to the internal resistance in the cool walls. On the other hand, if the heat units to be tested give off a part of their heat by radiation and these radiators are so placed that this radiant heat strikes the cool walls, it is then essential that the walls be designed with an average amount of internal heat resistance, in order that the radiant heat which strikes the walls may build up the surface temperatures and obtain the same effectiveness as would be the case under practical conditions.

It is evident that a test room cannot be built which will meet every condition in practice, but one should be constructed which will meet average conditions. The first type of test room is flexible, for testing different sizes of heaters in the same room. It does not seem to be well adapted to testing heaters for temperature distribution, but is reasonably consistent for determining the output by the condensation method. The second type of room is not readily adapted to different sizes of heaters. For convection-type heaters, the walls may be of any construction and cooled by any method so long as the inner surfaces present a practical surface at the proper temperatures. For radiant heaters, the cold walls must have a practical amount of internal heat resistance in order that the radiant heat may be effective. With these precautions, it may be used to advantage for determining heat distribution on the heat output by the condensation method.

The next question which has always been a disturbing factor in considering the problem of rating radiators is that of a proper method to measure the output of the radiator even after a test room has been selected. The most common and easily applied method has been to measure the output by the condensation method; that is, the radiators have been tested, the total condensation under given conditions noted, and the rating determined by converting this condensation into British thermal units. This method is easily applied and gives the total amount of steam required of the heating plant, but does not show accurately the results which may be expected from a given type of heater.

Since there is admittedly a marked difference in the heat distribution from various types of heaters, it seems that this should at least be recorded by an acceptable test method. For this purpose, the first type of test room does not

seem to be the best selection, and especially not unless very great care is given to the method of interchanging the air in the room. If this type of room is to be used, the interchange should be effected in such manner as not to disturb the temperature gradient from top to bottom of the room as set up by the natural convection currents of the radiator itself.

Taking everything into consideration, the best type of test room would appear to be one in which the proper amount of outside wall surface is cooled and the walls are constructed with an average amount of internal heat resistance in order to give the radiant heat an opportunity to build up the surface temperatures and thus to be effective as a heating medium. Further than this, it would seem that the room temperature should be measured at a height of 30 in. from the floor line, at which point 68 F should be maintained, and the other temperatures from top to bottom should be recorded as a matter of showing the effectiveness of heat distribution from the particular radiator under test. There is undoubtedly a need for further research work to determine the proper type of test room and the proper test methods in order to take into consideration all of the important factors in connection with a heating unit.

CALCULATION OF HEAT LOSSES FROM BUILDINGS

The fourth and most important factor in determining heating plant requirements is that of calculating the heat losses from a building. In fact, this may be considered as the main part of the problem and as including in part the other three subdivisions as previously discussed. The major factors to be considered in this part of the problem are as follows:

1. Air conditions required within the building.
2. Outside weather conditions to be expected.
3. Thermal properties of the walls used in the construction.
4. Air leakage or infiltration through the various parts of the building.
5. The quality of workmanship which may be obtained during construction.

The complete information regarding these five factors has not, in the past, been at the disposal of the designer. This has led to much uncertainty, and, in many cases, the use of large and variable factors of safety. The lack of information on certain parts of the problem has served to cast a suspicion and doubt on data obtained for certain other parts and upon methods of calculation which appear scientifically correct, but for which certain necessary factors or coefficients have not been available.

There are two general paths or ways by which heat may escape from a building.

1. By direct transmission through the walls of the structure.

2. By infiltration or an interchange of air from the inside to the outside of the building.

The laws governing the flow of heat by direct transmission have had a great deal of study, and it is now possible to calculate with accuracy the heat flow through most types of walls under conditions of constant temperature difference between the air on the two sides of the wall. Under this condition, the transmission is governed by the temperatures and velocities of the air on the two sides of the wall, by the internal heat resistance of the wall, and, to a certain extent, by the temperatures of the surrounding objects. When these

conditions remain constant, it is comparatively easy to determine the demand on the heating plant. If, however, there are sudden changes in the air temperatures, the demand on the heating plant may not immediately follow these changes, due to the heat capacity of the wall.

Consider two walls, one built of brick 24 in. thick, and the other built of cork 1½ in. thick. These walls have practically the same thermal resistance. The first wall has a heat capacity of approximately 40 Btu per square foot of wall per degree drop in temperature, and the second wall has about 0.4 Btu of heat capacity under the same condition.

If a drop of 10 deg in the outside temperature is assumed, the additional heat flow through each wall after uniform conditions are reached would be $0.2 \times 10 = 2.0$ Btu per hour. The total heat storage of the brick wall after the changed conditions were reached would be 40×5 or 200 Btu less than before the temperature drop. It is evident that a wall of this type would be very slow in reaching normal conditions of heat flow and that the 200 Btu would act as a heat storage to relieve the maximum demand on the plant. The cork wall, on the other hand, would have only 2 Btu of reserve capacity and could not cause any such delay or lag in the heating plant requirements. A glass window would cause practically no lag in the maximum requirements.

This heat lag is caused by the stored-up heat in the wall and depends upon its heat capacity. The effectiveness of this heat capacity depends upon the conductivity of the wall. If the conductivity is low, it will last for a longer period of time than it will if the conductivity is high. The extent to which this heat capacity will affect the maximum requirements of the heating plant will depend not only upon the capacity and the conductivity of the wall, but upon the length of time during which the minimum outside temperatures prevail. It would seem that it can best be taken into account by a thorough study of the thermal properties of walls, together with an analysis of weather bureau records. From this analysis, outside temperatures may be selected for determining the maximum heating plant requirements which will apply to various types of construction. The necessity of considering thermal capacity of the structure as an element in calculating maximum heating plant requirements has been recognized by the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS and an extended research program is under way at the Research Laboratory.

Wind velocity affects the direct heat transmission through a wall by changing the surface coefficient. From Table 1 it is evident that the effect is

TABLE 1. COMPARISON BETWEEN WALLS IN FIGS. 1 AND 2 FOR STILL AIR AND A 35-MPH WIND VELOCITY

Wall	f° Still Air	f° 35-Mile Wind	Temp. Drop Air to Air	U Still Air	U 35-Mile Wind	Percent Increase in U
Fig. 1	1.5	8.0	70	0.15	0.163	8.67
Fig. 2	1.5	8.0	70	0.75	1.26	68.0

entirely different for walls of different internal heat resistances. No single factor can be obtained which will fit the various wall constructions and correlate wind velocity to temperature differences.

The logical procedure for determining heating plant requirements for walls subject to direct transmission appears to be as follows:

1. Select the inside air temperature to be maintained.

2. Consider the outside weather conditions to be expected. This includes the minimum temperature, length of time these temperatures prevail, maximum wind velocities, and direction to be expected.

3. Consider the thermal properties of the structure and select proper outside temperatures and conductivity coefficients to be used in the design of heating plant. From these data, calculate by well-known formulae the total amount of heat by direct transmission. Such a procedure needs no factor of safety or exposure factor, providing the proper care is used in selecting the coefficients.

The loss of heat by infiltration is an entirely different loss than that by direct transmission and requires a different treatment. The amount depends upon the size of the cracks around the windows and doors, the construction of the walls, the wind velocity and direction on the outside of the building, and, to a certain extent, the exposure. The losses due to infiltration have in the past been the most uncertain part of the problem, and have often been very high, especially under heavy wind conditions. This uncertainty has led to the use of exposure factors which, on the windward side of the building, have at times run as high as 40 per cent. Such exposure factors can only be justified by the lack of knowledge which has existed in regard to these losses. They are an admission that something has been omitted in the design, and this has usually been the effect of high wind velocities on increasing the air infiltration. Wind velocities have been assumed which are too low and the amount of air leakage into the building has been under-estimated. With data which are now available, it is possible to determine more accurately the losses by infiltration, and, therefore, there is less need to use exposure factors.

CONCLUSIONS

In conclusion, consider the five specific parts of the problem as previously outlined for determining the heat loss from buildings:

1. The air conditions required within the building relate largely to air temperatures unless ventilation is required in which the heat loss through this channel should be considered as a separate part. With heating units which give good distribution of temperature from floor to ceiling, it should be reasonable to assume 70 F as the average temperature of the air within the room. If, however, heating units are to be used which give variations of 20 or more degrees from floor to ceiling, it will then be necessary to use higher average wall temperatures and especially higher temperatures for ceilings which are exposed.

2. The outside air conditions to be expected constitute one of the most difficult parts of the problem. This includes the minimum temperatures, wind velocities, the effect of sunshine on the building, temperature changes, and exposure. The minimum temperature and the direction and velocity of the wind may be obtained from the Weather Bureau records for most cases. The most difficult part of the problem, however, is to determine the maximum combined effect of temperature and wind velocity for any one period. As previously pointed out, wind velocity affects heat transmission through the wall by changing the outside surface coefficient, which may make a rather high

percentage of differences for poorly insulated structures. Wind velocity has a direct effect on infiltration, and for high wind velocities, the infiltration losses of poorly constructed buildings may be greater than the direct heat transmission losses. Complete information on this subject should give the maximum wind velocity and direction to be found for any temperature condition during the heating season. It would then be possible for the designer to determine the maximum requirements for the building. The effects of sunshine and temperature change may be of less importance in determining the maximum load on the heating plant than they are for determining the maximum load on the cooling plants for summer conditions. They are, however, very important factors in determining maximum requirements. They must be considered in conjunction with heat capacity of the structure as previously explained.

3. The heat transmission, through various types of construction, has received a great deal of attention and a great deal of data are available, covering different types of wall construction. It is true that the greater part of the coefficients have been considered from the point of constant temperature differences, whereas the maximum load on the heating plant may be greatly affected by the heat capacity of the wall and the duration of low temperatures to be expected. This part of the problem needs more thought and research. A consistent study of the weather bureau records is also necessary in order to determine the proper outside temperatures to use with walls of different thermal properties.

4. The air infiltration losses have probably caused more failures and have been more difficult to determine than any other part of the problem, except, perhaps, the outside weather conditions to be expected. This is natural, as there are so many factors involved. With proper construction, however, these losses may be greatly reduced, and, from data now available, may be very closely determined. The data which are now most needed may be obtained from a comprehensive analysis of the weather bureau records to find the wind velocity and direction which may be expected at different temperature ranges. Losses can then be accurately determined, but any failure to determine them cannot be scientifically taken care of by an exposure factor, as there is no logical reason for applying such an exposure factor on one side of the building more than on any other, excepting for the reason that proper coefficients were not selected.

5. The quality of construction which may be expected, may play a very important part in the heat losses. It will affect both the direct transmission losses and the air leakage losses. The most logical method of allowing for this seems to be to take it into consideration when selecting the transmission coefficients and infiltration factors. If any factor of uncertainty were to be applied to the calculations, this would seem to be the most logical point of application.

Wind velocities and temperature gradients have two distinct effects on the amount of heat loss from a building. Wind velocity affects transmission by changing the outside surface coefficient of the wall. For well insulated walls, it will have but little effect, while for thin, highly conductive walls, it may have a very appreciable effect. Temperature differences affect the transmission in direct proportion to the differences. There is no relation between the two which will fit all cases.

Take as an illustration the two walls of Figs. 1 and 2. For reasonably still air, the outside surface coefficients are 1.5, and for wind velocities of 35 mph, they will be about 8.0. Under these conditions, the conductivities for a temperature drop of 70 F to zero, would be as shown in Table 1. From these results, it is evident that no factor will correlate wind velocities and temperature differences.

The factors which at present seem to require more information in the problem of calculating maximum heating plant requirements are: *first,* a more comprehensive study of weather bureau records; *second,* a definite recognition of the heat capacity of walls.

DISCUSSION

PRESIDENT CARRIER: This paper is simply the essentials of an orderly, comprehensive outline of a subject that could be elaborated for many years of study by this Society.

I am particularly interested in the heat capacity of the walls and its effect in averaging up the variations in outside temperature to give a wall effect which is more nearly an average daily or sometimes weekly outside temperature variation. This is true not only in heating but in cooling and air conditioning practice. We are making a research along those lines and we want the ideas of every one in the Society regarding it.

E. K. CAMPBELL: Our experience shows the necessity of taking into account the heating-up load of a cold building which is intermittently heated.

In a test at Butler University the building was heated for a period of 12 hrs starting at — 5 F. We found that about 42 per cent of all the heat put into the building was absorbed by the interior material, and that the heating-up load in an intermittently heated building is an important factor. It is responsible for many heating plants which are calculated in the ordinary way failing to give satisfactory service because they do not have the proper capacity.

A heating plant located in the South, calculated according to the extreme low temperatures on the same basis as a heating plant in the North, does the work in severe weather much more quickly than the Northern plant, because the cold spells in the South do not last long enough for the cold to penetrate the walls of the building.

W. L. FLEISHER: The practical application of the paper to the actual work of the installing companies should be considered. Because of inefficiency of workmanship, standards were developed originally by the heating contractor. It is difficult to figure the efficiency of construction and consequently there should be established a standard of good construction through which a certain number of heat units can leak. On that basis the standards of the Society should be set up in order to protect the engineering fraternity.

W. A. SWAIN: One of the basic factors of all heating and effective heat in a building is dependent on outside conditions. If the Society indicated now to the Weather Bureau a weather curve that fulfilled the needs of the industry and the Society and have the curve published regularly such valuable statistics would be available when needed in the future.

PROF. A. P. KRATZ: Professor Rowley has made a thorough and timely analysis of all the factors that exist in the problem, and from that standpoint the paper appeals to me as being a classic of common sense.

E. C. EVANS: About 14 years ago the *American Institute of Architects* had a committee on building construction that later on was to have worked with our Research Laboratory so that construction could be standardized with assurance of low heat losses. I do not recall that there has ever been a report or reference to the work of that committee since that time.

Perhaps the Chair or the Secretary could tell us of this committee and of our relations with them to the end that buildings will be designed with more thought given to their heat loss.

If Professor Rowley's classic is to come to fruition, we must have such relations with the *American Institute of Architects* so that our research will be used by them.

PROFESSOR ROWLEY: Mr. Evans, I do not know of anything that is being done in an official way. There is, I believe, close cooperation between the *A. I. A.* and this Society. I believe the architects are accepting the various coefficients as given in THE GUIDE and are endeavoring to get the best type of construction possible. As soon as the public in general becomes educated as to what is really required and as to the inefficiency of poor construction, the problem will be simplified.

No. 905

STEAM CONDENSATION AN INVERSE INDEX OF HEATING EFFECT

By A. P. KRATZ [1] (MEMBER), AND M. K. FAHNESTOCK [2] (NON-MEMBER),

This paper is the result of research conducted at the University of Illinois in cooperation with the A. S. H. V. E. Research Laboratory.

The data presented in this paper were obtained in connection with an investigation conducted by the Engineering Experiment Station of the University of Illinois, of which Dean M. S. Ketchum is the Director. This work, which has been in progress for the past five years, is a part, and a continuation of a general research program devoted to the study of Heating Rooms by Direct Steam and Hot Water Radiators, and carried on in the Department of Mechanical Engineering under the general supervision of A. C. Willard, professor of heating and ventilation and head of the department. Acknowledgment is due to E. L. Broderick, research assistant in mechanical engineering, for valuable assistance in the collection and preparation of the data.

THE object of this part of the investigation was to make a comparison of the performance of several types of direct cast-iron radiators using steam as the heating medium, when the temperature of the air in the test room was maintained at approximately 70 F at the breathing level, with the performance of the same types of radiators when the temperature was maintained at 68 F at a level 30 in. from the floor. It was suggested that such a comparison might justify recommendations for the adoption of a standard for testing such radiators.

DESCRIPTION OF APPARATUS

All tests were conducted in the low temperature testing plant which has been completely described in previous publications.[3] In brief, this plant, as shown in Figs. 1 and 2, consists of two rooms erected within a larger room having 6-in. cork walls, floor and ceiling. The latter room is equipped with direct expansion refrigerating coils, and each of the two tests rooms presents two walls of standard frame construction exposed to any desired temperature

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[3] University of Illinois Engineering Experiment Station Bulletins Nos. 192 and 223. Also Transactions of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, Vol. 35, 1929, pages 79-89.

Presented at the Semi-annual Meeting of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, Swampscott, Mass., June, 1931, by A. P. Kratz.

maintained in the cold room. One wall of each test room contains a pair of double hung windows and the other wall a standard door with a glass panel. In addition, arrangements are made so that the floors and ceilings of the test

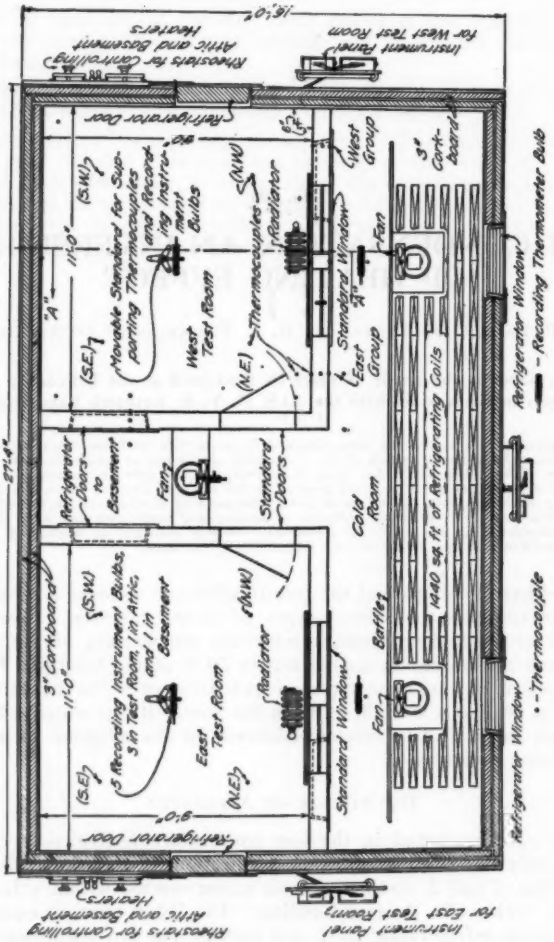

FIG. 1. PLAN SECTION OF LOW TEMPERATURE TESTING PLANT

rooms may be exposed to any desired temperatures. Air movement over the walls is obtained by means of fans.

The plant is thoroughly equipped with thermocouples and temperature recorders in order to permit all observations being made without the necessity for entering the rooms and thus disturbing conditions at any time during the

period of a test. The radiators are connected with a single pipe for steam and return condensation, and the condensation is weighed as shown in Fig. 2. A steam separator is installed in the line to each radiator, and an electrical super-

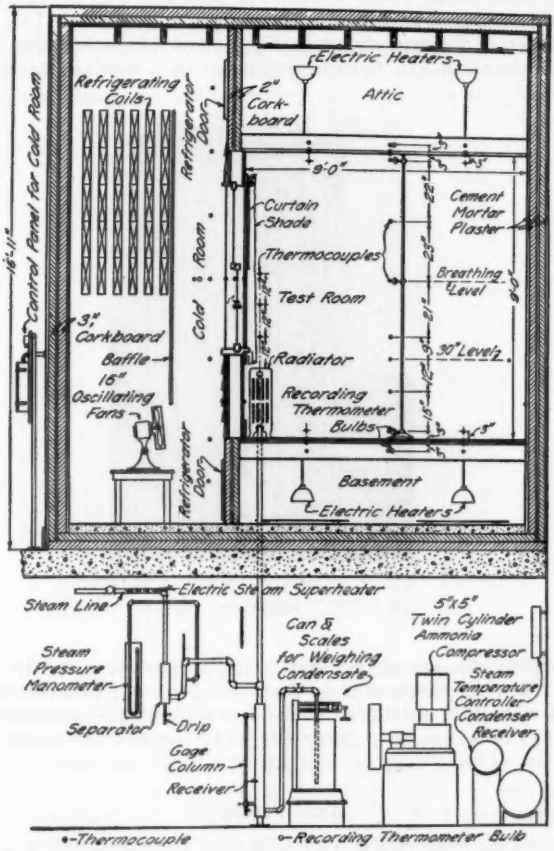

FIG. 2. ELEVATION SECTION OF LOW TEMPERATURE TESTING PLANT

heater further insures the use of dry steam. Provision is also made for venting the radiators to prevent the accumulation of air.

The temperatures at the various levels within the test rooms are observed by means of thermocouples made from No. 22 B and S gage copper and constantan wire and supported on a standard in the center of the room as shown in Fig. 2. A study of temperature conditions within the test rooms proved

that the center reading at a given level is representative of the average temperature at that level.

The cast-iron radiators used for the present series of tests consisted of (1) a 6-section, 26-in., 5-tube radiator, as shown in Fig. 3; (2) a 10-section, 26-in., 3-tube radiator, as shown in Fig. 4; (3) a 13-section, 17-in., single tube panel radiator of the type shown in Fig. 5; and (4) a 3-section wall type of radiator as shown in Fig. 6, each section having a nominal rating of 7 sq ft.

FIG. 3. 6-SECTION, 26-IN., 5-TUBE RADIATOR

More complete dimensions and details for setting are given in the inset in Fig. 8. When the temperature at the 30-in. level was maintained at 68 F as compared with 70 F maintained at the 5 ft level, it was necessary to increase radiator (1) to 9 sections, radiator (2) to 12 sections, and radiator (3) to 16 sections, but no increase in radiator (4) was found necessary.

TEST PROCEDURE

In all cases the temperature in the cold room was maintained at about —2.5 F and one of the exposed walls of the test room was subjected to an equivalent wind velocity of approximately 10 mph. The temperature of the air above the ceiling of the test room was maintained at 62 F and the temperature of the air 3 in. below the floor about 2 deg higher than that 3 in. above the floor. No test observations were made until conditions had remained constant for several hours, as indicated by the readings of the thermocouples on the inside wall surfaces. This required about 20 hr after a change in radiators was made. When the required thermal constancy had been attained, the condensate was weighed over a period of one hour, and no test was accepted if the condensate showed more than 2½ per cent deviation in the successive 10-min increments

of weight. At the end of each test a separate test was run to determine the condensation in the piping alone, and the total condensate was corrected by subtracting the amount so determined.

RESULTS OF TESTS

The results of the tests for which the temperature of the air at the breathing level (5 ft above the floor) was maintained at approximately 69 F are shown in Fig. 7. From these curves it is evident that while the radiators exhibited

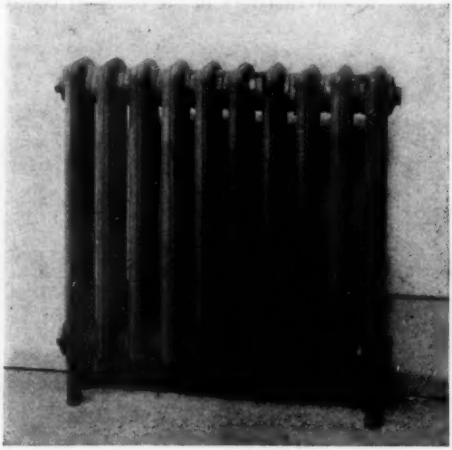

FIG. 4. 10-SECTION, 26-IN., 3-TUBE RADIATOR

wide differences in characteristics relative to temperature distribution within the test room, the steam condensation required to maintain 69 F at the breathing level was practically the same for all of the radiators. The maximum deviation in condensation was only 3.5 per cent.

Wide deviations in the degree of comfort produced in the room by the different radiators were apparent. With the exception of the wall type radiator, when the radiators were operated to maintain 69 F at the breathing level, the room was decidedly too cool for comfort. Some discomfort undoubtedly resulted from the fact that the inside surface temperature of the exposed walls was approximately 55 F, but in the case of the panel and the two tubular radiators, by far the greatest amount of discomfort was caused by the low temperatures in the zones below the breathing level, particularly in the zone from the 30-in. level down to the floor.

The differences between the steam condensations were not great enough to justify an attempt to evaluate the practical *heating effect* of the different radiators in terms of the steam condensation alone. The greatest temperature difference from floor to ceiling and the lowest temperatures between the floor and the 30-in. level were obtained with the 5-tube radiator. In this case the

temperature at the floor was only 53.8 F and at the 30-in. level was only 59.9 F. The room was most uncomfortable when this radiator was used. The temperature at the breathing level obtained with this radiator was approximately one degree lower than the 69 F obtained with the other radiators. A slight increase in the size, sufficient to bring the breathing level temperature to 69 F would have increased the condensation slightly and increased the temperatures at all levels in the room by approximately one degree, but would not have appreciably increased the comfort. Thus, while the positions of the curves representing the temperatures at the levels below the breathing level afford a means of ranking the different radiators in the order of comfort produced, or

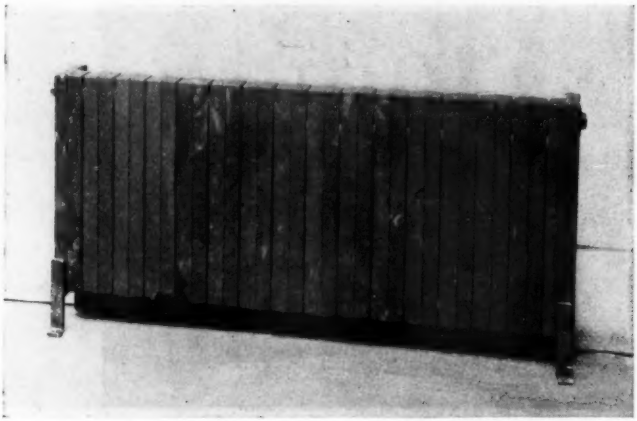

FIG. 5. 16-SECTION, 17-IN., 1-TUBE PANEL RADIATOR

heating effect, they constitute an intangible and rather unsatisfactory measure inasmuch as the statement of the results is entirely qualitative and not quantitative.

The 3-section wall type of radiator was the only one that produced a satisfactory degree of comfort in the room with the breathing level at 69 F. The reason for this, as previously mentioned, was that the temperature in all zones below the breathing level were higher than those for the other radiators. It was particularly noted (see Fig. 7) that the temperature at the 30-in. level was approximately 68 F. This indicated a possibility that if the sizes of the other types of radiators used were increased by amounts necessary to raise the temperature at the 30-in. level to 68 F in every case, that each radiator would maintain practically the same degree of comfort in the room, and a tangible measure of the heating effect might be obtained from the increase in condensation resulting from the increase in size necessary to obtain the 68 F at the 30-in. level.

The results of the tests with the air at the 30-in. level maintained at approximately 68 F are shown in Fig. 8. In all cases the room was comfortable and it is at once evident that the magnitudes of the steam condensations arrange

themselves in the same order that radiators would be ranked from an inspection of the room temperature gradient curves. The total spread between steam condensations was increased to 21.5 per cent as compared with the 3.5 per cent obtained with the air at the breathing level maintained at 69 F. The 5-tube radiator, with the greatest temperature difference between the floor and the ceiling, also gave the greatest steam condensation, while the 3-section wall type, with the least temperature difference, gave the least condensation. For the other radiators, the condensation was also in the order of the temperature difference. From Table 1 it is also evident that a consistent reduction in

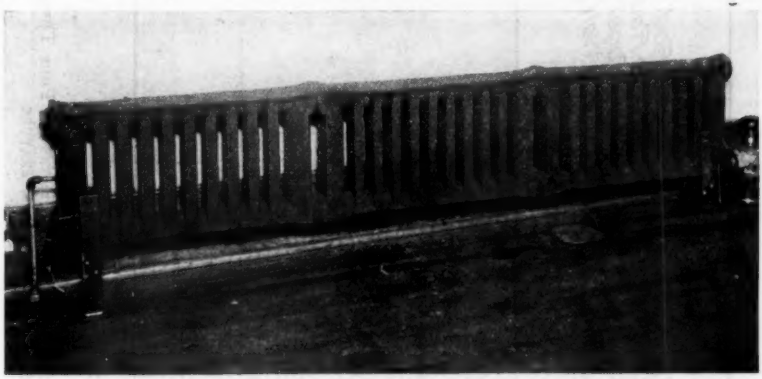

FIG. 6. 3-SECTION WALL RADIATOR

condensation occurred with a reduction in difference in temperature between the floor and ceiling.

The temperature at the 30-in. level obtained with the wall type radiator was approximately one degree Fahrenheit lower than that for the other radiators. A slight increase in the size of the radiator, sufficient to raise the temperature at the 30-in. level one degree Fahrenheit would have increased the temperature at all levels by the same amount, thus giving a new curve parallel to the old one and one degree higher. Hence, the temperature difference between the

TABLE 1. COMPARISON OF STEAM CONDENSATIONS AND TEMPERATURE DIFFERENCES BETWEEN FLOOR AND CEILING FOR A CONSTANT AIR TEMPERATURE OF 68 F AT THE 30-IN. LEVEL

Type of Radiator	Temperature Difference Ceil.-Floor (F)	Steam Condensation (Lb per Hr)	Ratio of Each Value in Columns (2) and (3) to Preceding Value	
			Temperature Difference	Steam Condensation
(1)	(2)	(3)	(4)	(5)
5-tube	21.6	6.68	....	....
3-tube	17.5	6.48	0.810	0.970
1-tube panel	13.8	6.12	0.788	0.943
3-section wall	10.8	5.50	0.782	0.900

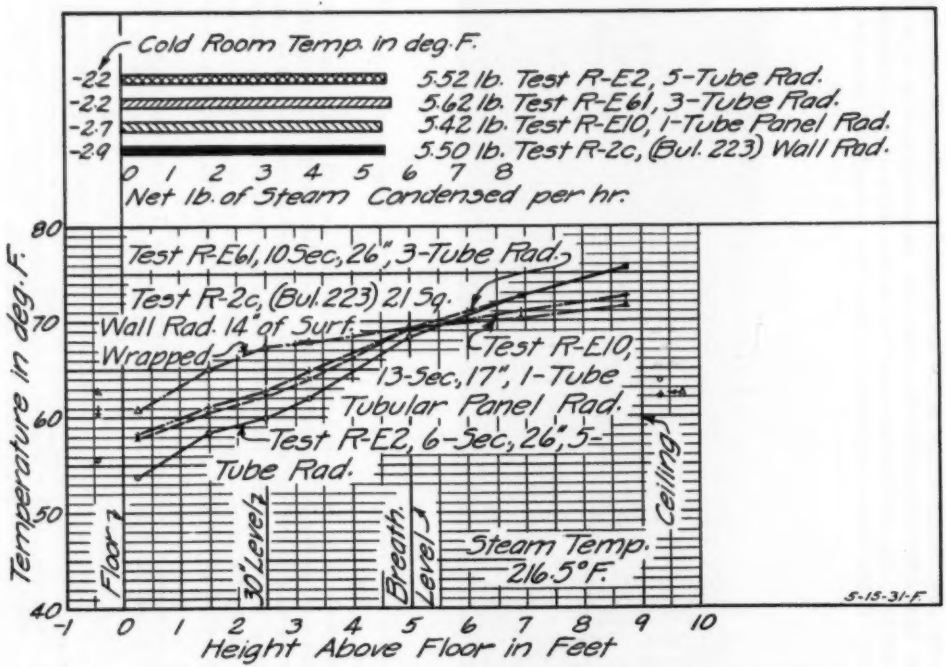
Cold Room Temp. in deg. F.
-2.2
-2.2
-2.7
-2.9
5.52 lb. Test R-E2, 5-Tube Rad.
5.62 lb. Test R-E61, 3-Tube Rad.
5.42 lb. Test R-E10, 1-Tube Panel Rad.
5.50 lb. Test R-2c, (Bul. 223) Wall Rad.
0 1 2 3 4 5 6 7 8
Net lb. of Steam Condensed per hr.
Temperature in deg. F.
80
70
60
50
40
Test R-E61, 10 Sec., 26", 3-Tube Rad.
Test R-2c, (Bul. 223) 21 Sq. Ft. Wall Rad. 14" of Surf. Wrapped
Test R-E10, 13-Sec., 17", 1-Tube Tubular Panel Rad.
Test R-E2, 6-Sec., 26", 5-Tube Rad.
Floor
30" Level
Breath. Level
Ceiling
Steam Temp. 216.5° F.
5-15-31-F.
-1 0 1 2 3 4 5 6 7 8 9 10
Height Above Floor in Feet

FIG. 7. ROOM TEMPERATURE GRADIENTS AND STEAM CONDENSING RATES FOR FOUR TYPES OF C. I. RADIATORS WITH A COMMON 60-IN. LEVEL TEMPERATURE

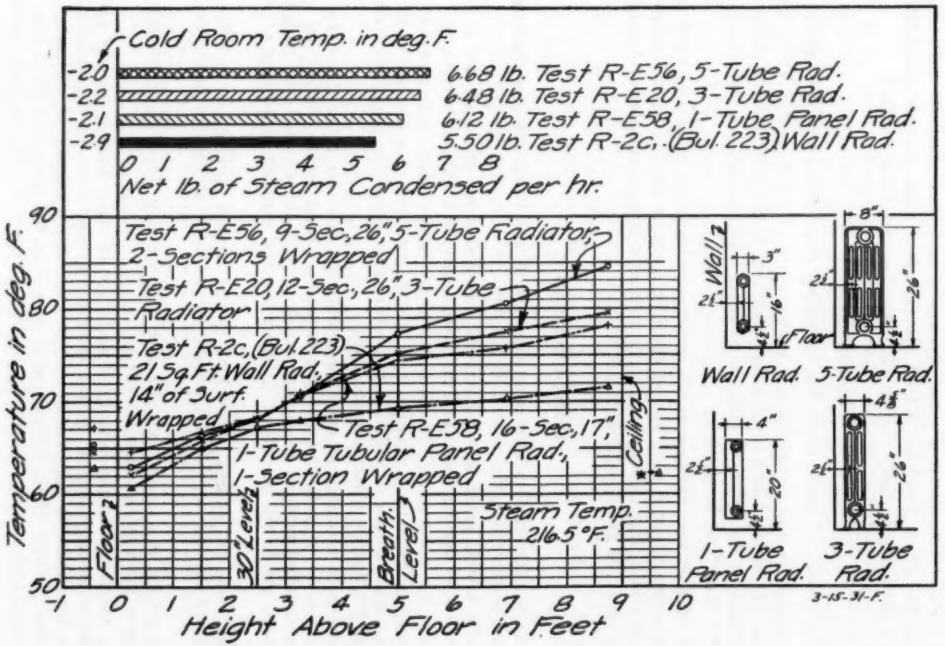

FIG. 8. ROOM TEMPERATURE GRADIENTS AND STEAM CONDENSING RATES FOR FOUR TYPES OF C. I. RADIATORS WITH A COMMON 30-IN. LEVEL TEMPERATURE

floor and ceiling would remain the same and the ratio of temperature differences shown in the last figure of column 4 in Table 1 would not change. The steam condensation, however, would increase slightly over the 5.50 shown in column 3, thus increasing the ratio of condensations shown as the last figure in column 5 and making it more nearly in conformity with the next to the last figure, or 0.943. Therefore, if exact agreement had been obtained in the temperature at the 30-in. level for all of the radiators even greater quantitative consistency than that actually shown in Table 1 would have resulted.

CONCLUSIONS

From the results of these tests, the following conclusions may be drawn:

1. It is apparent that maintaining a temperature of 68 F at the 30-in. level is a more acceptable standard for stating conditions under which direct radiators shall be tested in the type of plant used than maintaining a temperature of 70 F at the breathing level.

2. With 68 F at the 30-in. level, the steam condensation obtained in the test plant used seems to be a measure of the effectiveness of the direct radiators tested, in providing for human comfort, or, more briefly, condensation becomes an inverse index of *heating effect.*

DISCUSSION

S. R. LEWIS (WRITTEN): This is an exceedingly interesting paper characterized by the usual concentrated efficiency of the authors.

The results in this paper bear out my contention that we are ready for a new measuring stick in heating, analogous in a rough way to a Watt in electricity.

I suggested the name *Kent.* The Kentage of any steam heated heat transmitting medium would be the ratio of its heat input to the heat input of a standard radiator such as the 5-tube radiator described in this paper, when producing equal temperature 30 in. above the floor in a standard test room.

It should not be necessary to use an elaborate test room or artificial cooling of that room in order to make this simple comparison. Referring to Table 1 we can compute by standard methods the approximately correct amount of 5-tube radiation for any room, and it will be difficult to learn any new methods of arriving at this figure.

Every day, however, we meet a new kind of heat transfer surface, and in different rooms in every building we find it expedient to use different types of heat transfer surface.

We have arrived at a time in our education when we must admit that condensation is not the true direct index of heating effect.

I propose therefore that the Kentage of each type of steam heating transfer agent shall be determined at any of several laboratories accredited by the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, at the expense of the manufacturer and that Kentage (I do not care what we call it, but some new short term is desirable) shall begin now to be used by heating engineers in selecting heat transfer agencies.

For instance, referring to Table 1, the Kentages are as follows:

	%
3-tube radiator, 26 in. high	103
1-tube panel radiator, 17 in. high	106
Wall radiator, 13 in. high	111

For example, we compute the total heat loss for a room with all usual factors for transmission, infiltration, exposure, etc., and find that we need 100 sq ft of 5-tube radiation, this having been rated on the basis of ¼ lb of steam input per square foot per hour, a total of 25,000 Btu per hour.

We propose to heat the room with wall radiation 13 in. high, the Kentage of which is 111. We therefore need in this case to install but 90 sq ft of the wall radiation.

$$\frac{100^k}{111k} = 90$$

If we had the Kentage each of the various types of concealed convectors, and of variously located and speeded unit heating machines, much grief would be avoided.

The Kentage of each type of heater is to be determined in a standard test room in a laboratory.

The Kent is the ratio between the heat input of the heater required to maintain 70 F at a point 30 in. above the center of the test room floor to the heat input of a 38-in. 3-tube free-standing, cast-iron radiator of such size as will maintain 70 F at a point 30 in. above the center of the test room floor.

Thus we can compute the heat loss from a room in the ordinary way and then divide this by the Btu input required at the heater, as is the custom, and then when we select the heater we may multiply the required room input in Btu by the Kentage for a convector type heater having whatever draft head we may select; or by the Kentage of the unit heating machine we may select, and by so doing may take credit for the cooler ceiling or other characteristics which may pertain to the heater which we select.

The manufacturer of heater X would rate his heater as one having an *input* of say, 100,000 Btu per hour, Kentage 130 per cent.

If I had a room which figured 120,000 Btu I would, if I took the X heater, pick the 100,000 Btu input size because it would be ample for the 120,000 Btu input room, allowing for its higher efficiency as a transfer agent.

S. S. SANFORD: I note that the most satisfactory radiator was the wall type radiator which was placed nearest the floor, that is, the distance from the floor to the top of the most satisfactory radiator was 16 in. The distance to the top of the next most satisfactory radiator was 20 in. and the distance to the top of the least satisfactory radiator was 26 in. I am wondering if the wall type radiator had been placed so that its top was 26 in. from the floor if it still would be the most satisfactory radiator?

PROF. F. B. ROWLEY: One question I should like to ask of Professor Kratz, in regard to the relation between average room temperatures and condensation is as follows. In Fig. 7 where the condensations are practically the same, isn't it true that the average room temperatures are about the same? There is a much wider range of air temperatures for the high radiator from floor to ceiling but the average temperature is substantially the same as for the low radiator giving about the same amount of condensation. In Fig. 8 the wall type of radiator shows a much more uniform distribution of air temperature from floor to ceiling with a lower average temperature and also a lower condensation rate. In other words, the condensation is about proportional to the average air temperatures of the room.

A radiator which throws a large proportion of its heat directly to the top of the room without means for diffusion or bringing it down to the floor level is ineffective. It appears that the high type of radiator such as the 38-in., cast-iron radiator performs in this manner. The air current leaves the top of the radiator at a higher temperature and therefore at a correspondingly higher velocity than for the lower type of radiator. These hot air currents reach the top of the room and remain there

unless there is some artificial means of diffusion to bring them down to the floor. This means uneven heat distribution with an excessive heat at the ceiling where it is not required. An effective radiator should keep the heat at the lower part of the room, either by diffusing it from the radiator at low temperatures, by diffusing it under windows where it is mixed with the cold draft coming down from the glass or by diffusing it through radiation to the lower parts of the room. A radiator to be effective must give uniform heat distribution. After high-temperature air is once thrown to the top of the room it cannot be brought down except by some artificial means of diffusion.

It seems to me that the result as obtained in the researches at the University of Illinois and as brought out in Professor Kratz's paper are of extreme value and have a practical application. Certainly from these it is possible to select radiators which will be effective in heat distribution and therefore efficient in operation.

One thing in connection with Mr. Lewis' definition of his proposed *Kent*—if such a definition or term were to be adopted, I believe it would be better to compare the radiator with a perfect radiator or one which gives perfect heat distribution and not to compare it with a 38-in., 3-column radiator which we know gives imperfect results and probably will be disregarded in the near future.

R. C. BOLSINGER: Did Professor Kratz figure out the percentage of the radiant effect in proportion to the envelope of the radiator, that is, the outside surface between the worst radiator, the 26-in. radiator, and the wall radiator? One of the reasons for the better heating effect of the wall radiator is that on the first test the wall radiator was about 20 to 25 per cent longer. In that way the surface is lengthened and there is obtained not only the benefit of the additional amount of air to be moved over that surface, but also the additional benefit on account of the envelope of the surface from the radiant effect of that radiator which, in the final analysis, is the reason for the heating effect, by dissipating the heat at the low floor level, and with the lower temperature at the ceiling the heat balance loss between the outside and inside air is not as great.

E. HARRINGTON (Schenectady): In Table 1 it appears that the condensation is some function of difference in temperature between floor and ceiling. In other words, there is a saving of about 20 per cent in condensation by reducing the temperature between floor and ceiling from 21 to 10. We have found in testing forced convection electric heaters that in a typical test room we can get a differential between floor and ceiling of from 2 to 3 F. I assume the same thing would apply to unit heaters for forced convection steam radiators. Would not the saving in condensation continue to go on down with this low temperature differential and perhaps be on the order of 30 per cent or more?

F. D. MENSING: As far as application is concerned, I think we are working on something that does not exist. The tests might be applicable to basements or ground floor rooms. According to Professor Kratz, he endeavored to maintain the space below his test room at a temperature as close to that immediately above the floor as possible. Now it so happens in our building construction, except in the basement, the room below is always warmer at the ceiling than at the floor. This paper brings that out. With a room below heated the same as the room tested, there would be temperatures of 76 and 61 as against those shown of 72 and 54. In other words, in the particular room tested there is a heat flow downward and we always have the heat flow upward as the ceiling below is warmer than the floor above. In the next table the figures become 85 and 72 as against 65 and 61. I have never found earth temperatures in basements below 54 F. What we have really tested is a basement room or a ground room on something similar to concrete or some other similar material. Even a concrete floor in a building is warmer than the temperatures that they had in this test room.

J. F. McINTIRE: I do not agree that we should give more consideration to the temperature 30 in. above the floor. As a matter of fact, we are interested more in the temperature at the floor for comfort. Professor Kratz in his office sits 8, 10 or 12 ft away from a wall radiator and is comfortable. Therefore he is interested in this subject of the heating effect of the room and the amount of condensate produced by that radiator, but in the Metropolitan Life Building, which is up for consideration now, the architects have seen fit to specify a sheet metal shield mounted 1 in. in front of the radiator to prevent the radiant heat from striking the employees because they are going to place desks in every available square foot of that office to make it pay.

Therefore, in that building the principles brought out in this paper do not apply but they do apply in Professor Kratz's office where there is plenty of room and he can sit away from the radiator. It is not the time to rate radiators and adopt a Kent or any other term until we analyze the application and consider what the radiator is going to be required to do.

PROFESSOR ROWLEY: I heartily agree with Mr. McIntire, but I do feel that these tests are certainly worthwhile and they show exactly what is going on in many of the radiators in our homes. Regardless of whether or not you put on a shield we are using them right along under these conditions, and the problem is to get a comfortable temperature in the living zone. Now if one type of radiator gives a temperature much higher at the ceiling, and a high range of temperature, we certainly should know it and there is no better way of finding it out than to put these radiators in a test room and try them out, as Professor Kratz has done, and then select radiators which will give comfortable living zone conditions. Certainly there is a difference between radiators giving variations of 25 F from floor to ceiling, while others will come down to 5 F. No doubt radiators or heaters of some kind can be designed which will give practically uniform conditions from floor to ceiling and resulting in efficient operation. Of course there cannot be obtained colder conditions at the ceiling than at the floor, but the room can be heated and radiation losses prevented. There are two losses: one due to the high temperature at the ceiling and the other due to loss of radiant heat.

MR. BOLSINGER: In taking exception to Mr. McIntire on the question of the radiant effect, I believe the shield would give an ideal effect from the radiating source because resulting radiant energy would be mild and at a lower temperature. If that radiator were designed so that it was low and covered a large surface, and both the convection and the mild radiant effect were had then I think the ideal effect mentioned by Professor Rowley would be obtained.

MR. MENSING: Some real tests should be made with the floor below the test room having the same temperature and similar conditions as exist in the test room. Tests should be made under the same conditions as we have in our homes, and in our offices. A person who has worked or lived in a room where it is cold underneath, in spite of all tests and no matter what the temperature may be at any height, knows that the cold space below makes his feet cold. In the final analysis we are selling comfort, not heat.

PROFESSOR KRATZ: So far as Mr. Lewis' written discussion is concerned, I think his idea is an excellent one. Maybe we can work toward getting a unit to express these things. However, with the amount of test data that we have at present, I do not believe we can commit ourselves to any such unit. In the opening statement of his paper he said it would not be necessary to run tests. I believe it necessary to run a large number of tests, even to establish the Kentage of a given radiator.

Mr. Sanford asked something about what would happen if we raised the wall-type radiator. When we started out we also thought that the important factor was the

height of the radiator; that the higher the radiator the worse the air conditions would be. To our surprise we find that the thickness factor seems to have more bearing on this problem than either the height or the length. The thick radiators give the greatest temperature difference between the floor and the ceiling, while the thin radiators give the least temperature difference. A year ago I would have answered your question unqualifiedly by saying if you raised the radiator you would get greater temperature difference between the floor and the ceiling. Now I am not so sure of it.

Professor Rowley's question about the average temperature of the room being the same when the condensation was the same bears out an observation. When we made all the radiators maintain 70 F at the breathing level, the temperature gradient curves seemed to swing themselves around that point and to give practically the same average temperature in the room, but we observed that when we made them maintain 68 F at the 30-in. level, there was no tendency to swing around a point and give the same average temperature in the room. In other words, it differentiated for the different radiators between the average temperature in the room and therefore reflected the effect in the steam condensation.

Mr. Bolsinger mentioned the proportion of radiation or the relation between the radiation and convection effect. In one of our earlier papers we attributed the better heating effect of the long, low wall radiator to the very thing that he points out. In that paper I think we said that the low radiator had a larger percentage of direct radiating surface and because the direct radiating surface was placed lower better comfort was obtained.

Mr. Harrington suggested that by still further reducing the temperature gradient between floor and ceiling we would obtain still greater economy in the use of the radiators. Our results point that way. Whether there is a limit to that or not, I cannot answer definitely, but certainly the less temperature difference existing between the floor and the ceiling the greater will be the economy, but probably as the small temperature differences are reached the curve will flatten, thus preventing enormous reductions in steam condensation.

Mr. Mensing states that we have only run these tests in one room that simulates a first floor room. Certainly that is all we have done. It is impossible to run tests in all types of rooms at once when limited by time, apparatus and staff. So up to the present we have concentrated on this particular type of test, letting the room represent a first floor room. Ultimately, we expect to try to do some other things.

No. 906

EXPERIMENTS TO DETERMINE HEATING EFFECT FACTOR

By R. N. TRANE[1] AND C. J. SCANLAN,[2] LACROSSE, WIS.
MEMBERS

CONVECTION-TYPE heaters are rated by most manufacturers in terms of square feet of equivalent direct "radiation" (240 Btu per hour) and the performance of these heaters usually is compared with that of exposed or direct cast-iron "radiation." However, shortly after the introduction of this type of heating equipment, it became apparent that equivalent temperatures could be maintained within the living zone of a room with less condensate when properly designed convection heaters were used than when exposed cast-iron "radiation" was used. This reduction in heat input to do the same heating work has been termed *heating effect.*

This difference was accounted for by the fact that floor temperatures were higher and ceiling temperatures were lower with convection heaters than with direct heaters. Although this claim was in a measure substantiated, it was impossible entirely to account for the large difference in condensation requirements between a good convection heater and an exposed cast-iron "radiator" by means of the small temperature differences existing within the room itself. Thus, although a properly designed convection heater usually will deliver a greater quantity of the available heat to the living zone of a room, temperature differences alone apparently do not entirely explain this *heating effect.*

Convection heaters, as well as exposed cast-iron "radiators" are now rated in terms of the square foot of equivalent direct "radiation" (240 Btu per hour) based on condensation tests run in still air at a temperature of 70 F. A convection heater must of necessity be entirely different from an exposed cast-iron "radiator," both structurally and physically.

Although an exposed cast-iron "radiator" may actually emit 240 Btu per hour per square foot of surface under standard conditions of 215 F (steam) and 70 F (air), the most efficient convection heaters have a very small amount

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Presented at the Semi-annual Meeting of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, Swampscott, Mass., June, 1931, by R. N. Trane.

of prime surface and a great deal of extended surface, and the heat emission therefore has no such relation to the actual heating surface.

Efforts have been made to combine *heating effect* with the rating of a convection heater. A code for testing and rating convection heaters for both steam and hot water is now under preparation by a committee of the Society. This code, when completed, should provide an accurate and correct basis for determining the heat output of a convection heater at the outlet grille. The important question, however, is—*What becomes of the heat delivered from this outlet grille in an actual installation?*

PURPOSE OF INVESTIGATION

One purpose of the investigation reported in this paper was to determine why convection heaters require less heat input to maintain the same air temperature within the room than exposed cast-iron "radiators" when the room is exposed to identical cooling conditions in both cases. Another was to ascertain whether *heating effect* should be combined with rating or whether it is a function of some physical factor. In other words, the primary object was properly to evaluate the Btu input to a convection heater as compared with that of the various exposed cast-iron "radiators."

LABORATORY EQUIPMENT

In order to accomplish this purpose, it was necessary to build a cold room of a type adaptable to the desired testing methods. A study was made of the cold room used at the University of Illinois, and also at the Frost Research Laboratories. In considering a cold room similar to the type used at the University of Illinois, the infiltration through windows and doors seemed to complicate the problem, in addition to the difficulty of maintaining similar conditions for a number of tests. It was therefore decided to build a room which eliminated infiltration and one having walls that could be cooled to any desired temperature and maintained at that temperature over an indefinite period. In other words, it was necessary that this room be so constructed and equipped that constant temperatures could be maintained for a period of eight hours or more if desired. Thus, infiltration and variable temperatures were eliminated, although it was realized that in practice infiltration cannot be disregarded.

The cold room used contained a heating space 8 ft wide by 12 ft long by 8 ft 3 in. high. Three walls were constructed of sheet metal and were painted on the interior the same as interior walls would be painted. Four thermocouples were soldered on the surface of each of these walls so as to obtain surface temperatures with rapidity and accuracy. A thermocouple was located 3 in. above the floor, at the breathing line, and 3 in. below the ceiling, approximately in the center of the room. A thermocouple was also placed immediately over the location of the heater at a distance of 3 in. from the ceiling and 3 in. from the wall. Two thermocouples were installed 2 in. above the floor and opposite the inlet to the heater, and two thermocouples were also provided for measuring the outlet temperature from the heater. The system was constructed in such a way that all thermocouples were balanced. All lead wires were made

the same length and from the same spool of wire. The readings were taken by means of a milivolt meter.

A five-ton CO_2 compressor was located in the room beneath the laboratory, and was used for cooling water in an 800-gal tank. This water was in turn pumped through the nozzles in the spray chambers behind each of the sheet metal walls.

Figs. 1 to 7 illustrate the equipment used.

PROCEDURE

Thirty-eight tests of 8 hr duration were conducted to determine the operating peculiarities and characteristics of the equipment before attempting to establish

FIG. 1. FRONT OF COLD ROOM

the proper procedure for operating the cold room. It was desired to ascertain whether or not stabilized conditions could be maintained over a period of hours. It was found that a point of equilibrium could be obtained when the heat input to the room exactly balanced the heat output.

The equilibrium was obtained in all cases by adjusting the size of the exposed cast-iron "radiator" or convection heater to the heating requirements. A smaller exposed cast-iron "radiator" was required with a 62-F wall temperature than was required with a 54-F wall temperature. Likewise, shorter convection heaters were required with a 62-F wall temperature than with a 54-F wall temperature.

After this preliminary work had been completed, it was decided that in order to obtain a comparison of the relative heating effects of the various exposed

TABLE 1. RESULTS OF TESTS

Test Number	Equipment Tested	Floor Temp.	Breathing Line Temp.	Ceiling Temp.	Ave. Living Zone Temp.	Ave. Room Temp.	Ave. Wall Temp.	Air Temp. Entering Heater	Air Temp. Leaving Heater	Difference Floor to		Pounds Condensate	Btu
										Breathing Line	Ceiling		
1	5 Tube—38 in.—N. and W. Walls	63.5	72	79.6	68.0	71.8	54	73.8	163.1	8.5	16.1	4.34	4200
1	" N. and W. "	65.0	72	79.0	68.0	72.0	58	70.0	169.0	7.0	14.0	3.69	3580
1	" S. and W. "	67.2	72	76.2	69.4	72.0	62	72.8	169.4	4.8	9.0	2.50	2425
2	Convection Heater *A*	65.0	72	75.0	69.0	71.0	58	71.0	148.0	7.0	10.0	2.47	2400
2	" " "	67.0	72	74.8	69.6	71.3	62	73.0	128.0	5.0	7.8	1.94	1880
2	" " "	69.0	72	74.3	70.2	71.6	66	75.7	142.6	3.0	5.3	1.10	1067
3	Convection Heater *B*	63.6	72	76.5	67.8	70.7	54	69.5	120.0	8.4	12.9	3.39	3285
3	" " "	66.0	72	76.0	69.0	71.2	58	75.0	106.8	6.0	10.0	2.56	2480
3	" " "	66.0	72	75.2	69.0	71.1	62	75.6	101.2	6.0	9.2	1.61	1560
3	" " "	67.1	72	75.2	70.0	71.5	66	79.0	95.5	4.9	8.1	1.07	1038
4	Convection Heater *C*	63.2	72	75.0	67.7	70.1	54	68.2	130.6	8.8	11.8	3.30	3200
4	" " "	65.8	72	76.5	68.9	71.4	58	71.0	124.8	6.2	10.7	2.51	2420
4	" " "	68.3	72	74.0	70.1	71.3	62	75.8	128.7	3.7	5.7	1.56	1515
4	" " "	69.0	72	74.8	70.5	71.9	66	76.8	104.0	3.0	5.8	1.18	1140
5	5 Tube—26 in. Heater	65.5	72	77.2	68.6	71.5	54	72.1	145.0	6.5	9.7	4.10	3980
5	" "	65.8	72	77.1	69.1	71.8	58	73.0	180.0	6.2	11.3	3.58	3470
5	" "	67.0	72	77.0	69.5	72.0	62	75.0	173.0	5.0	10.0	2.95	2860
6	Convection Heater *D*	64.3	72	75.9	68.1	70.7	54	68.8	136.6	7.7	11.6	3.41	3300
6	" " "	65.1	72	75.3	68.3	70.6	58	69.2	130.5	6.9	10.2	3.01	2920
6	" " "	66.7	72	75.0	69.4	71.3	62	72.2	118.0	5.3	8.3	1.91	1850
6	" " "	68.7	72	74.1	70.3	71.6	66	75.0	129.0	3.3	5.4	1.03	1000
7	5*B*—Wall Section—4 Sections...	65.8	72	75.0	68.8	70.9	54	77.4	146.5	6.2	9.2	4.38	4250
7	" " " ...	66.8	72	74.7	69.2	71.1	58	79.1	146.4	5.2	7.9	3.40	3300
7	" " " ...	67.8	72	73.7	69.8	71.1	62	79.3	159.5	4.2	5.9	2.35	2280
7	" " " ...	68.8	72	74.2	70.4	71.7	66	75.6	151.0	3.2	5.4	1.75	1700

cast-iron "radiator" and convection heater tests it would be necessary to establish a definite temperature at some point within the room. A breathing line temperature of 72 F was therefore selected. It was also decided to maintain a definite cooling effect in each test, and one of the side walls and the end wall were selected for this purpose. These two cold-wall surfaces comprised an area of 166 sq ft, and several tests were conducted with surfaces maintained at temperatures of 54 F, 58 F, 62 F and 66 F.

These wall surface temperatures were selected because they would give results comparable to outside temperatures ranging from —20 F to +30 F according to the results obtained at the University of Illinois Research Residence, as reported in the paper *Wall Surface Temperatures* by A. C. Willard and A. P. Kratz.[3]

TEST RESULTS

The test results are given in Table 1. Attention is called to the fact that constant breathing line temperatures were maintained throughout the investigation, and that the wall temperatures were varied. All of the tests reported in Table 1 are for a cold wall having a surface area of 166 sq ft.

DISCUSSION OF RESULTS

Because of the fact that equilibrium was maintained during all of the tests, it follows that the heat input to the cold room equals the heat loss. Therefore, inasmuch as the cold surface remained the same in all cases, not only in area but in temperature, and the inner air temperatures remained the same, the only factor of the room that could change was the coefficient of transmission of the wall (U). Fig. 8 indicates that with a wall surface temperature of 58 F, an exposed cast-iron "radiator" required an input of 3440 Btu per hour to maintain a breathing line temperature of 72 F. Therefore:

U_{c1} = Overall coefficient of heat transmission of wall for cast-iron "radiator"

$$= \frac{3440}{166 \times (71 - 58)} = 1.59$$ Btu per hour per square foot per degree difference between the wall temperature and the average room temperature.[4]

For a similar wall surface temperature, the convection heater required an input of 2400 Btu per hour. Therefore:

U_c = Overall coefficient of heat transmission for convection heater

$$= \frac{2400}{166 \times (71 - 58)} = 1.11$$ Btu per hour per square foot per degree difference between the wall temperature and the average room temperature.

[3] See A.S.H.V.E. TRANSACTIONS, Vol. 36, 1930.

[4] The overall coefficient of heat transmission for the test wall meets the definition of surface coefficient. For test conditions the wall surface temperature is also the temperature to which heat flows. In practice the walls have greater resistance than the sheet metal wall tested, but this does not change the fact that it is the overall coefficient of heat transmission (U) that is affected by the source of heat supply. The test wall at 58 F with a cast-iron "radiator" passes 20.7 Btu per square foot per hour, which checks a standard wall having a coefficient of transmission (U) of 0.25 Btu per square foot per hour per degree which for —10 F to 70 F gives 20 Btu per square foot per hour heat loss. The convection heater with a 58 F wall passes 14.49 Btu per square foot of wall area.

By means of these coefficients, the heat input required can accurately be predetermined. The *heating effect* is therefore equal to

$$\frac{U_c}{U_c^1} = \frac{1.11}{1.59} = 0.7$$

A comparison of floor-line and breathing-line differences, or floor-line and ceiling differences, for the exposed cast-iron "radiator" and the convection heater tested cannot possibly account for this difference in heat input required to maintain the cold room at 72 F.

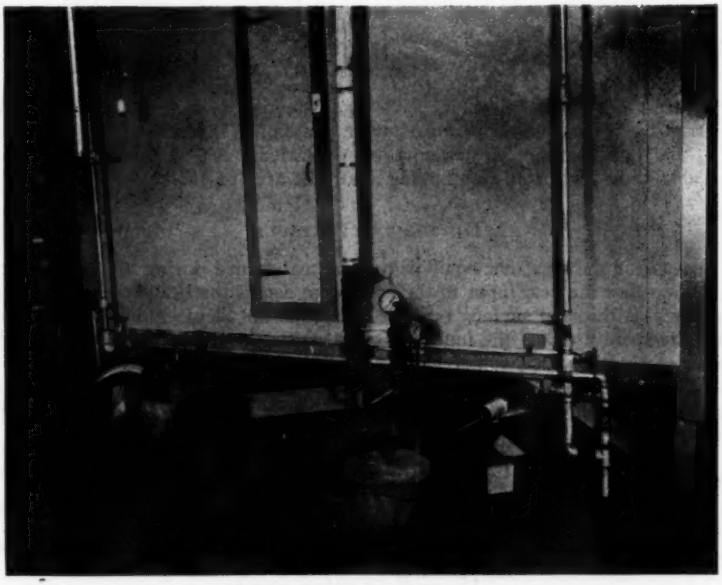

FIG. 2. REAR OF COLD ROOM

The apparent reason for this reduction in heat input required by the convection heater compared with that of the exposed cast-iron "radiator" is that in one case the heat emitted from the exposed cast-iron "radiator" is made up of convected heat and radiant energy, while in the other the heat emission is almost purely convected heat. Inasmuch as convected heat only is capable of raising the air temperature, it follows that in the case of the particular room tested the percentage of radiant energy emitted by the exposed cast-iron "radiator" was 30 per cent as compared to the convection heater tested. No two convection heaters of different design will have the same *heating effect* coefficient.

Fig. 8 shows average results obtained for all of the tests. No material difference could be determined between *heating effect* coefficients for high and

low exposed "radiators." Likewise, no appreciable difference could be detected between a convection heater with a high stack and one with a low stack. The only restriction in all cases is that of controlling the heat output of the convection heater or exposed cast-iron "radiator" to exactly equal the heat loss from the room.

It may be difficult to appreciate that these percentages are as large as herein reported. However, in Fig. 11 of a paper entitled Determination of Radiant

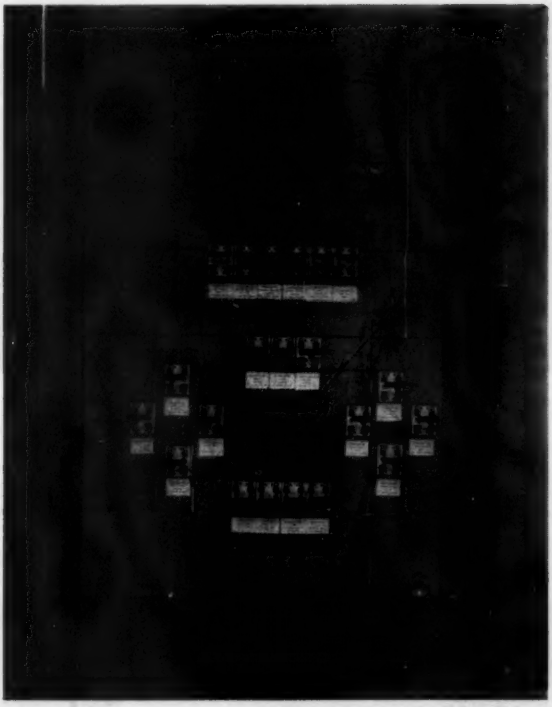

FIG. 3. THREE THERMOCOUPLE SWITCHBOARD

Energy Given Off by a Direct Radiator, by John R. Allen and Frank B. Rowley[5], the following percentages are given:

No. 1—Plain Cast-Iron—Rusty	38.3 per cent
No. 2—Aluminum Paint	25.0 per cent
No. 3—Black Paint	35.4 per cent
No. 4—Plain Cast-Iron—No Rust	31.2 per cent

[5] See A. S. H. V. E. TRANSACTIONS, Vol. 26, 1920.

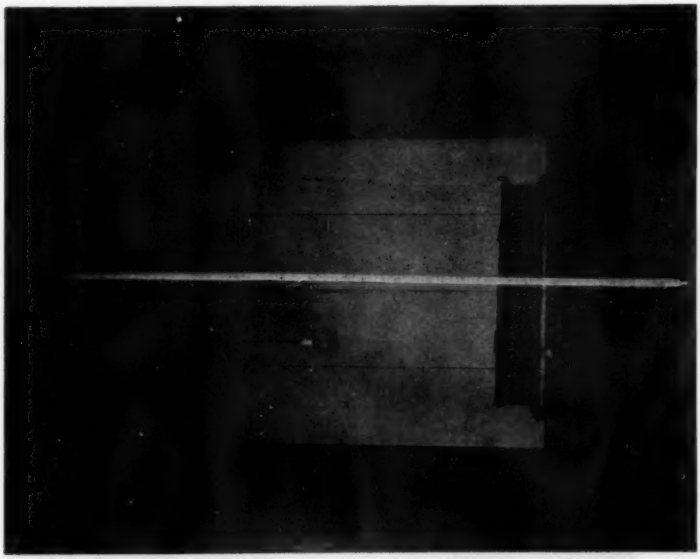

FIG. 5. SPRAY CHAMBER

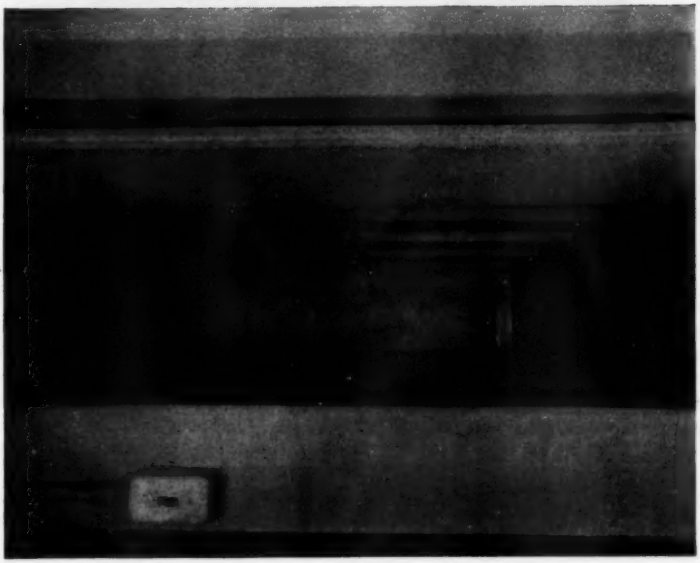

FIG. 4. INTERIOR OF COLD ROOM

Bulletin No. 120 (March, 1921) entitled Investigation of Warm Air Furnaces and Heating Systems, published by the University of Illinois Experiment Station, Fig. 43-A, on page 114, shows that coefficient of emissivity for bright tin, not insulated, to be 1.28 Btu per square foot per hour per degree Fahrenheit; whereas, for black iron, No. 28 U.S.S. gage, very rusty, it is 2.37. Inasmuch as the surfaces of both materials were subjected to the same steam and air temperatures, it follows directly that for the black iron specimen 46 per cent of the heat was given off in the form of radiant energy.

Figs. 23 and 25 of University of Illinois Experiment Station Bulletin No. 223 (March, 1931), indicate the results of tests with practically the same breathing-level temperature. If the test on enclosure No. 3 (test E-la) is compared with the test on unenclosed "radiator" (test R-5b), the *heating effect* will be found to be 0.855. Also, in Fig. 25 of this bulletin, the *heating effect* factor for test E-3a as compared with test R-5b is 0.862. The heaters referred to in these tests are primarily radiators and are as a result poor convectors. Stream line air flow and proper heating surface contacts have been given little consideration. If, as reported in these tests, the addition of a cabinet or enclosure produces a factor of approximately 0.85, it follows that a good convector, scientifically designed, will give without question the *heating effect* factor of 0.7 herein reported. However, this factor will not be the same for convectors of different design.

In order to illustrate the manner in which this *heating effect* coefficient is used in the practical application to heat loss calculations, the following problem, involving a warm air installation with no radiant energy[6], is submitted:

Calculated heat loss (0-70 F)	140,000	Btu
Infiltration loss	29,000	Btu
Wall and glass loss	111,000	Btu
Heating effect coefficient	0.7	
Wall and glass loss for convection heater....	77,700	Btu
Add infiltration loss	29,000	Btu
	106,700	

By test the average Btu input to this residence on a zero day was 110,000 Btu.

CONCLUSIONS

1. That the heat loss from any given room or building is dependent upon the source of heat supply.

2. That *heating effect* measures the change of the overall coefficient of heat transmission for a wall under the influence of different sources of heat.

3. That the *heating effect* coefficient applies only to wall and glass loss and not to infiltration loss.

[6] The data were taken from Bulletin No. 189, University of Illinois Experiment Station, Daily tests No. 105 and 111, page 26, corrected to a mean outdoor temperature of zero.

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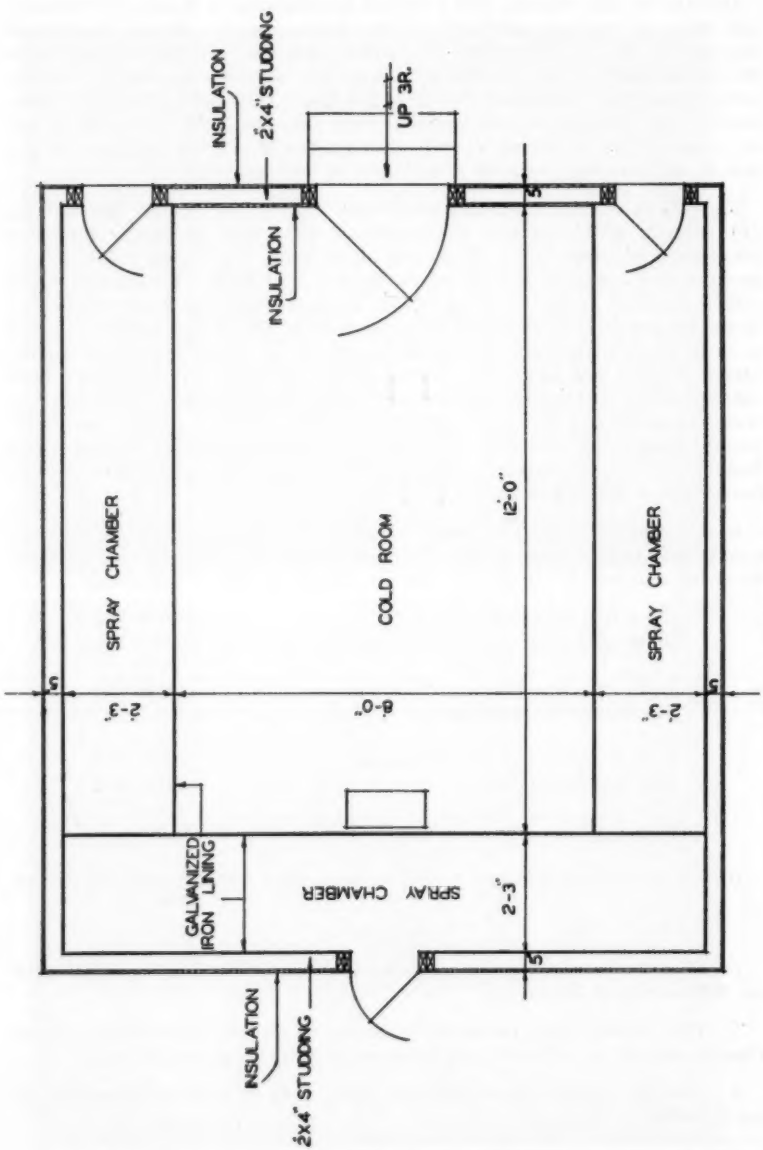

FIG. 6. PLAN OF COLD ROOM

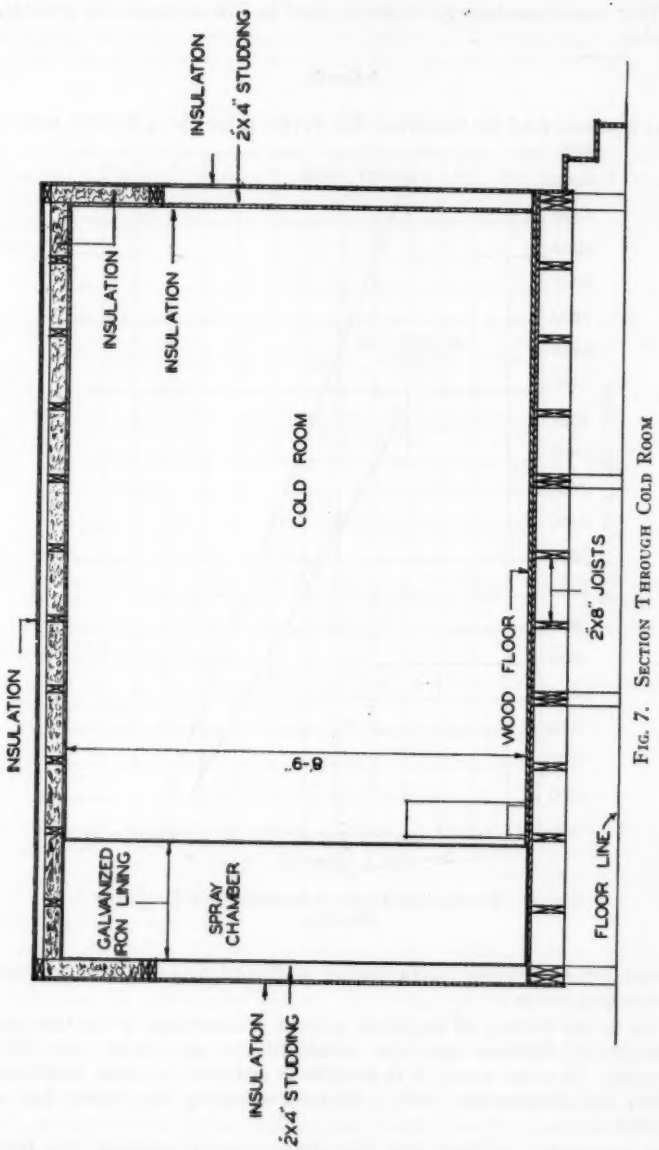

FIG. 7. SECTION THROUGH COLD ROOM

4. That convection heaters should be rated in Btu output at the outlet grille or opening.

Addenda

After completion of the data recorded in this paper the following tests were

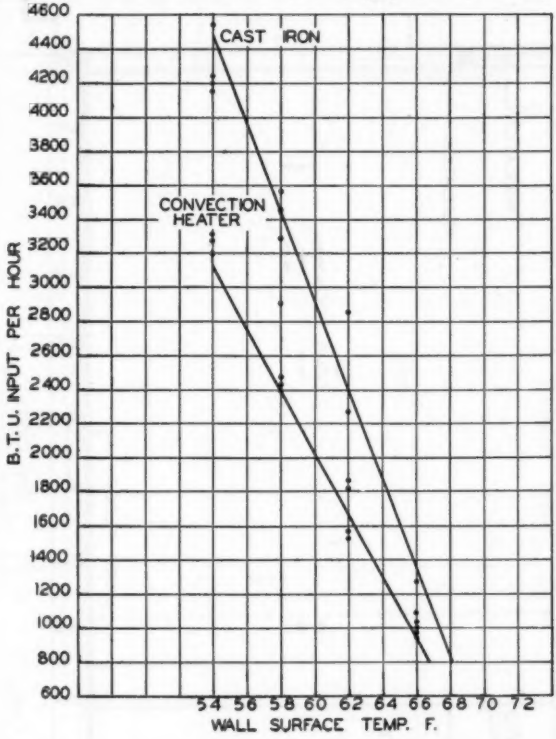

FIG. 8. RESULTS OF TESTS OF EXPOSED AND ENCLOSED HEATERS

conceived for the purpose of furnishing additional proof of the existence of radiant energy losses.

So far as the heating of the air in a room is concerned, it has been proved that heaters of different types can accomplish the same work with different heat inputs. In other words, it is possible to maintain the same floor-line and breathing-line temperatures with a heater condensing less steam than some other heater.

It is a generally accepted fact that radiant energy emission is a function of surface roughness and color. Therefore, highly polished surfaces are poor

radiant energy emitters and also poor radiant energy absorbers. Thus, by nickel plating the surface of the cast-iron "radiator," it is possible to eliminate the radiant energy emission and leave only the convection heat emission.

LABORATORY EQUIPMENT

The cold room, as described and illustrated in this paper, was used for determining the capacities of the plain cast-iron and nickel-plated sections. In addition, a set-up similar to that described in the proposed Code for Testing and Rating Concealed Gravity Type Radiation was used for comparative purposes.

PROCEDURE

The plain cast-iron section painted with gray lead paint was tested for a period of six hours in the same manner as described in this paper. In like manner, the section when nickel plated was tested for the same period. The hot room still air tests were run for a period of two hours each. During the cold room tests, the same breathing-line temperature—namely, 72 F—was maintained with both the nickel plated "radiator" and the painted "radiator" section. The same standard of heating perfection existed in both cases.

TEST RESULTS

The results of the tests are as follows:

Cold Room Test. A gray painted cast-iron "radiator" section condensed 1.47 lb of steam per hour. Same section (nickel-plated) condensed 0.71 lb of steam per hour. (*Note:* Wall temperature 66 F in both cases. Breathing-line temperature 72 F in both cases.)

Hot Room Test. Cast-iron section (painted) condensed 1.50 lb of steam per hour under standard-testing conditions.

The nickel-plated section condensed 0.75 lb of steam per hour.

DISCUSSION OF RESULTS

It is to be noticed that the results obtained in the cold room checked very closely with the results obtained in the hot room. It seems apparent from this that a measure of the output of a "radiator" can be obtained accurately by the hot-room method as provided for in the proposed code. This is also one of the conclusions given in this paper.

The ratio of condensation of the nickel-plated "radiator" (one section) to that of the painted "radiator" is 50 per cent. This factor is lower than that determined for the convection heater. This discrepancy between these two factors probably lies in the fact that the "radiators" originally tested were of more than one section, and, consequently, emitted a smaller percentage of radiant energy per square foot of rated surface.

The purpose of the test was to demonstrate by means of a known factor—namely, that a polished surface reduces the radiant energy emission—that the change in the overall coefficient of transmission of a wall (U) when convection heaters are used is due to the elimination of radiant energy effects.

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The results obtained check very closely those reported in Bulletin 120, University of Illinois Experiment Station, page 115, for bright tin and black iron cylinders.

CONCLUSIONS

That the difference in heat input required by a convection heater (nickel-plated cast-iron or non-ferrous) as compared to a cast-iron "radiator" to heat a room to equal standards of perfection is due to the elimination of radiant energy emission in the case of the convection heater.

DISCUSSION

A. C. WILLARD AND A. P. KRATZ (WRITTEN): Before proceeding to the discussion of the data contained in the paper presented by Messrs. Trane and Scanlan, a brief consideration of the principles involved in testing radiators in a closed room is not out of place. Such a closed room is essentially a calorimeter, and the total heat output of the radiator must be exactly equal to the heat loss from the room, if thermal equilibrium exists. Hence, if two different radiators maintain the same mean temperature for the air in the room, and exhibit a difference in steam condensation, or heat output from the radiator, the conclusion is inescapable that the heat loss from the room must be different in the two cases.

The change in heat loss from the room may result from a redistribution of temperatures at the various levels or from a change in the inside surface coefficient of heat transmission, as suggested in the paper under discussion, but in any case the change must be compensated for, either by a difference in size, or efficiency of heat transmission in the radiators, in order to produce the required difference in steam condensation. The difference in steam condensation directly reflects the change in heat loss from the room. Therefore, comparisons made on the basis of steam condensations obtained in a room in which the changes in heat losses under the given circumstances deviate widely from those that would occur in a room with practical standard construction under the same conditions, while interesting as applying directly to the room in which the tests were made, have no interest nor significance as applied to the actual rooms of standard wall construction in which the radiators must later be used.

It is obvious from the tests made by Messrs. Trane and Scanlan on radiators in a room with water cooled walls, that, at the same breathing level temperature for a given air to wall surface difference in temperature, there is a change in the *inside surface coefficients* as shown and computed in the paper.

Thus, the tests show:

$$f_{c.i.} = \frac{3440}{166\,(71-58)} = 1.59 \text{ Btu for a metal wall, heated by a bare cast-iron radiator.}$$

$$f_c = \frac{2400}{166\,(71-58)} = 1.11 \text{ Btu for a metal wall, heated by a convector.}$$

The heat lost from the room is determined by the overall coefficient of heat transmission, U, the complete expression for which is:

$$U = \frac{1}{\frac{1}{f_i} + \frac{1}{c} + \frac{1}{f_o}}$$

in which $\frac{1}{f_i}$ = resistance of inner surface of wall, including radiation and convection, and also, conduction of air film.

$\frac{1}{c}$ = resistance of the wall itself, where c = conductance.

$\frac{1}{f_s}$ = resistance at the outer surface of the wall, involving the conductance of the air or water film.

In the case of the water cooled wall, f_s is of the magnitude of 350[1] Btu per square foot per hour per degree Fahrenheit, thus making $\frac{1}{f_s}$ of the order of 0.00285. The thickness of the metal wall was not given, but it is hardly greater than ⅛ in. The conductivity, k, for sheet steel is of the magnitude of 400 Btu per square foot per hour per degree Fahrenheit per inch. Using these values, $\frac{1}{c} = \frac{0.125}{400} = 0.0003$. *Hence,* in the case of the water cooled wall both $\frac{1}{c}$ and $\frac{1}{f_s}$ could be safely neglected. However, including them for the sake of completeness, the overall heat transmission for the water cooled walls used in connection with the two radiators becomes

$$U_{c.i.} = \frac{1}{\frac{1}{1.59} + 0.00285 + 0.0003} = \frac{1}{0.63215} = 1.582$$

$$U_c = \frac{1}{\frac{1}{1.11} + 0.00285 + 0.0003} = \frac{1}{0.90415} = 1.106$$

$\frac{1.106}{1.582} = 0.699$, or 30.1 per cent decrease.

Using *precisely these same values of inside surface coefficients* for an actual wall, such as those of the room heating testing plant in the University of Illinois, the overall heat transmission values are:

$$U_{c.i.} = \frac{1}{\frac{1}{1.59} + \frac{1}{0.30} + \frac{1}{2.6}} = \frac{1}{4.34} = 0.230$$ Btu for an actual wall heated by a bare cast-iron radiator.

$$U_c = \frac{1}{\frac{1}{1.11} + \frac{1}{0.30} + \frac{1}{2.6}} = \frac{1}{4.62} = 0.216$$ Btu for an actual wall heated by a convector.

$\frac{0.216}{0.230} = 0.940$, or a decrease of 6.0 per cent.

The value 0.30 is the conductance, and the value 2.6 is the outside surface coefficient for the actual wall in the test plant at the University of Illinois, when the air to inside surface temperature difference is (71 − 58) or 13 F, which is the condition on which the Trane water cooled wall test plant was operated in the tests of the two different radiators.

Finally, since the decreases of 30.1 per cent in the Trane plant and 6.0 per cent in the University of Illinois plant in the overall heat transmission coefficients, when convectors are used in place of exposed cast-iron radiators in each plant, are almost exactly equal to the decreases in steam condensation in the two plants respectively, it is apparent that a relatively larger exposed cast-iron radiator had to be used in the plant with water cooled walls to maintain the same breathing level temperature and inside wall surface temperature in both plants.

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[1] Heat Transfer in Ammonia Condensers, Part III, Univ. of Ill. Eng. Exp. Sta. Bulletin 209, Fig. 19, page 35.

This shows the fallacy of using water cooled walls *where the sizes of the radiators are adjusted to maintain a fixed temperature at the breathing level,* and then drawing conclusions based on condensation alone without taking account of the adjustments in radiator size that must be made with rooms having walls with different heat resistances.

It is obvious that a change in the inside surface coefficient takes place with different radiators, but conclusions based on tests in a given room will be valid only when the ratio of surface resistance to total wall resistance is comparable with that found in actual service walls.

N. D. ADAMS (WRITTEN): During some experiments run at La Crosse in 1926 to determine the adaptability of cast-iron radiation to stack heating, it was noticeable that the condensation required to maintain a constant room temperature was less than that required by direct radiation, that is, exposed cast-iron radiator.

We could not understand at that time why this was so, and as we were interested in the problem of whether or not the cast-iron radiator was as efficient a convector as the so-called copper radiation, we did not look further for the cause.

The results of our experiments at that time indicated that the cast-iron radiator has approximately the same efficiency as the copper radiation when used as a convection heater; but due to the decreased weight of the copper radiation, ease with which it could be housed, and the fact that it gave less resistance to the flow of air than the cast iron, we decided to specify it for the particular job at hand. This installation has been in operation for three years, proving very satisfactory.

Quoting from Allison J. Thompson in Man, Nature and Civilization: "The so-called chaotic condition of nature is largely a conception of the human mind—for we may be sure that all of the various laws, forces and processes of nature are functioning with respect to each other, each in its own peculiar manner, with what may be regarded as an uncanny precision, wherever and whenever they come into conjunction with each other. The more we learn about the great scheme of nature and the inter-relation of its various elements the less of chaos is found."

To understand the inter-relationship of these various elements of nature and the various laws, forces, and processes by which they are governed is the task of our research work. We are to be congratulated that there are men among the members of this Society who are willing to give their personal time and learning, backed by the financial resources of the industries which they represent, to the end that we may give to the future generations facts concerning the operation of the forces of nature, which will increase their comfort and decrease waste effort which so many of us expend in combating, working in opposition to these forces. This paper shows a thoroughness of preparation, untiring effort to exclude errors, and conclusions which we believe are fundamental and have a practical application to the industry as a whole.

In discussing this paper it seems to me that there are three questions which should be asked. (1) What was the incentive force which caused these experiments to be made? (2) Were the conditions surrounding the experiments such that the conclusions may be taken as facts? (3) If these conclusions are facts, then how may we apply them to the practice of heating and ventilating?

Some of the incentives which have served as stimuli for reactions in human and animal life are ambition, pride, good-will, philanthropy, curiosity, and the fascination of adventure, exploration, and research, together with the glory of accomplishment.

Much of the research work which has been done by individual members of this organization has as its incentive, the desire of improvement of manufactured products, in an effort to gain leadership in some division of the industry. The men who have spent their time so generously have been rewarded with the glory of accomplishment.

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Another incentive is that there has been no time in the history of business enterprise when it was so necessary to conserve the natural resources and to eliminate waste, thus decreasing operating overhead.

As for the conditions under which the experiments were made, let us consider the cold room which was built in La Crosse in which these experiments were conducted. This particular laboratory was built, after spending considerable time in studying the results obtained from a predecessor in the University of Illinois. As we discuss the arrangement of this particular cold room let us keep in mind that Mr. Trane and Mr. Scanlan sought to determine the cause of one effect in particular and therefore wished to exclude any conditions which previous experiments had shown conclusively did not affect the problem at hand.

As we study the details of their room, we notice the omission of windows in the exposed wall, the absence of attic space or basement, and in particular, the construction of an exposed wall surface which is of sheet metal definitely eliminating the possibility of infiltration.

We read in the University of Illinois Bulletin No. 31, Investigation of Various Factors Affecting the Heating of Rooms with Direct Steam Radiators by A. C. Willard, A. P. Kratz, M. K. Fahnestock, and S. Konzo, several interesting conclusions.

Conclusion 18. The use of storm sash alone does not materially affect the room air temperature conditions, but did effect a gain in steam economy of approximately 11 per cent in the particular room tested.

Conclusion 19. The use of a storm door alone improves room air temperature conditions, and in the particular room tested effected a gain in steam economy of approximately 21 per cent.

Conclusion 20. The use of both storm door and storm sash improves room air temperature conditions and in the particular room tested effected a gain in steam economy of 31.7 per cent. The actual saving in general will be dependent on the ratio of window and door area to exposed wall area, and on the air-tightness of the storm door and sash.

It is hardly necessary to discuss further that the omission of windows in exposed wall surface, for this particular problem only eliminated one of the variables which they would have to allow for in each test. We now believe that there is a direct relationship between the economy of a heating system and the amount of infiltrated air; a constant based on the temperature differential.

Conclusion 21. Variations in basement and attic temperatures do not have a material effect on room air temperature conditions.

Looking at the cross-section of the LaCrosse cold room, we note that the ceiling and front wall were well insulated while the floor was of a regular house construction, double flooring, open below to the room temperature. The paper does not state whether or not a constant temperature was maintained in the space surrounding the cold room. We believe, however, that such was the case and that a constant transmission was maintained during the 8 hr of each test between the chamber and the surrounding space. If the outside room conditions were maintained constant for all types of radiation tested, then the elimination of attic, basement and a front wall construction can have no effect on the results obtained.

The paper states that the sheet metal wall construction was used in order to eliminate infiltration and thus simplify the problem. Is it necessary that the coefficient of transmission of the exposed walls (U) be the same for two different experiments? No two buildings have exactly the same exposed wall construction. The important factor is that, before starting tests, the coefficient of transmission of the chosen wall construction be definitely established. This we believe was done in this

case, as 38 tests of 8 hr duration each were conducted to determine the operating peculiarities and characteristics which would affect the operation of the cold room.

We note that during the tests, for each 8-hr period, the inside wall surface was maintained at a constant temperature. The temperatures used were those found in the experiments conducted by Prof. A. C. Willard and Prof. A. P. Kratz at Urbana.

Conclusion 24. The temperature of the inside surface of exposed standard frame walls varies from 67 to 59 F for walls not exposed to air movement, and from 65 to 53 F for walls exposed to an air movement of approximately 10 mph as the outdoor temperature ranges from 40 to — 10 F and the indoor temperature remains constant at 72.5 F.

It is interesting to note that the overall heat transmission of this construction approximates that of the standard wall used at Urbana; when the average indoor temperature was maintained at 71 F with a — 10 deg outside temperature. There was a heat loss of 20 Btu per square foot per hour.

We have referred to the 38 tests of 8-hr duration which were conducted to determine the proper procedure for operating the cold room during the standard tests.

In all experimental work there is a question of proper methods in obtaining data, number of tests taken, and duration periods of tests in order to eliminate possibilities of error that might be due to fluctuating conditions. The consistency of the results as shown in Table 1 would indicate that sufficient number of tests, time and care had been taken.

Should some aviator attempt to establish a new altitude record, standard instruments are required and conditions imposed as set down by a committee of the *American Aviation Society* at the Bureau of Standards, Washington, which must be followed in order that the aviator may receive credit for his efforts if successful.

Could we, as a Society, through our Research Committee, set up a code of standard rules and regulations to be followed? Then in the event that these rules and regulations did not apply to new work, the committee could be notified before the test was to be made and have a representative on hand to approve the conditions and procedure, and certify the results.

If we now accept the conclusions of this paper as facts, let us consider how they may be applied to the practice of heating. The illustration given for the cold room at Urbana shows a heat loss of 140,000 Btu, while the Btu required if a convection heater is used is stated as 106,700 or 76 per cent of the calculated heat loss. This makes a savings of 24 per cent in steam consumption. If this be true, then, the radiation of any steam heating system to be installed hereafter should either be nickel plated or otherwise treated to prevent the giving off of radiant energy or some type of convection heater should be used.

It is also possible that there are many buildings using cast-iron radiators today where it would pay the owners to investigate the possibilities of replacing the cast-iron radiators with convection heaters, or enclosing the present radiators to convert them into convectors, or to nickel plate them, in which case the surface may not be sufficient for the space requirements.

EXAMPLE

Total square feet of radiation in hotel building	30,000
Season heat demand per 100 sq ft	53,000 lb
Total season demand for the building	15,900,000 lb
Total season's cost at $1.00 per 1,000 lb	$15,900
Total cost to replace building with cast-iron radiation with a non-radiant type at 50¢ a square foot	$15,000
24 per cent saving in steam demand	$3,816
6 per cent interest on owner's investment	$900
Net saving per year	$2,916

In addition to the economy created by the use of the convection heaters, we cannot fail to notice the increase of comfort in the room due to the smaller differential between the floor and ceiling temperature.

R. V. FROST (WRITTEN): The paper by Messrs. Trane and Scanlan is a splendid contribution to the current research on heating effect.

In the choice of laboratory equipment it is to be noted that for this work there has been selected a type of room that permits the isolation and separate study of the various factors bearing upon heat transfer. In the writer's experience this method is much more effective than to attempt to study the causes for heating effect by the aid of a test room built to simulate actual structural practice. There is no sound reason why in our research upon effective heat we should stick to test equipment that permits nothing other than general conclusions when by the proper selection

FIG. A. TYPICAL ROOM WITH LARGE GLASS EXPOSURE

of equipment, studies can be made of specific characteristics which permit one to accurately analyze each individual heat transfer factor that enters into the phenomenon of space heating.

To do this it is not necessary to build a room that is inflexible in heat loss characteristics. A test room can be as easily built to permit tests with different wall constructions as to confine the tests to a single type of construction. Criticisms have been frequent against the water cooled wall type of test room but if those making such criticisms will take the trouble to weigh all the factors, the reasons for their objections will be found insignificant in the face of those in favor of the water cooled walls. In the conduct of radiator tests, the most severe heating condition must be reproduced under controlled conditions. Obviously the most severe heating condition is that of a glass enclosed room exposed to a cold driving rain.

A typical room of this type is illustrated. A water cooled wall test room reproduces such a condition. Radiation must be rated to take care of the worst condition. Any heating load less than the worst need not concern us unless we are making a specific study of wall construction, for if a radiator will suffice for the worst weather condition it will of course prove adequate for any other. Study of wall construction must not be confused with the separate study of the radiator. They are related but not interdependent.

Messrs. Trane and Scanlan have taken a step in the right direction by the elimination of infiltration in this study. As they continue they will find it essential to more specifically study other factors. However while infiltration as a factor has been eliminated, it is not to be ignored and requires a special investigation.

One criticism to be made of the Trane experiments is that too few measurements of temperature have been taken. If temperature readings had been taken in at least thirty more locations in the room, data could have been gathered that would give a very much more complete picture. In the writer's experimental work on heating effect he has found it quite essential to base the investigation upon a study of the floor to ceiling temperature gradient. When this is done and account is taken of the temperature range throughout the room the study of heating effect is quite simplified.

In the Trane tests a uniform temperature of 72 F at breathing line was selected. Had the temperature gradient been studied instead, the uniform breathing line temperature would not have appeared so desirable. Professor Willard has proposed that a uniform temperature at the 30-in. level be used in comparing radiators. This is a step in the right direction, but as a result of the tests in the writer's laboratory the average temperature in the zone from floor to 36 in. above floor has been demonstrated as the best for comparison. This zone, which has been designated as the floor zone, is the one occupied by a person in a sitting position—the shoulder height being at about the 36-in. level. This zone must be maintained at a comfortable temperature if the occupants of the room are to enjoy genuine heating comfort. A person standing does not require as much heat as one sitting, so that to judge the heating of a room by the comfort of a person in a standing position is far from correct. Of two radiators, each giving 72 F at breathing line, one may cause a very much cooler temperature in the zone below the 36-in. level and thus be a less desirable radiator from the comfort standpoint. (See chart with temperature gradients of three different types of radiators.) On the other hand, of two radiators providing the same degree of comfort in the floor zone one may produce a temperature of say 4 F less at the breathing line. To compare radiators by the breathing line temperature only is a practice that cannot, in the light of our present knowledge, be permitted to continue.

Likewise, to compare radiators strictly on the basis of temperature at the 30-in. level is incorrect, for some radiators particularly of the cabinet convector type with horizontal outlet, will give a high temperature at the 30-in. level but a sub-normal temperature at the floor level. The temperature gradient chart of the three different types of radiators shows how comparison on any one level is incorrect.

In the Trane tests the assumption has been made that heating effect is solely a function of radiation. Here too a study of the temperature gradients shows that such an assumption is not correct. Both convection and conduction are important factors and since a poor radiator is also both a poor conductor and a poor convector, it is quite easy to confuse the effect of one with that of the other. Much of the heat loss that we have attributed to radiation the writer has found instead to be caused by direct conduction from the air to the glass or wall surface, for much of the air coming in contact with the glass and wall surface does so solely through convection.

Messrs. Trane and Scanlan have chosen to designate the heating effect coefficient as a decimal, such as 0.7 or 0.85 etc. In the writer's work the complement of the

decimal has been used, namely, 0.30 or 0.15, and the term has been expressed as a percentage, 30 per cent or 15 per cent, etc.

There are many reasons why such a designation is more practical, a few of which can be named.

1. Those accustomed to speaking of heating effect have formed the habit of saying one radiator has a better or poorer heating effect than another. Such an expres-

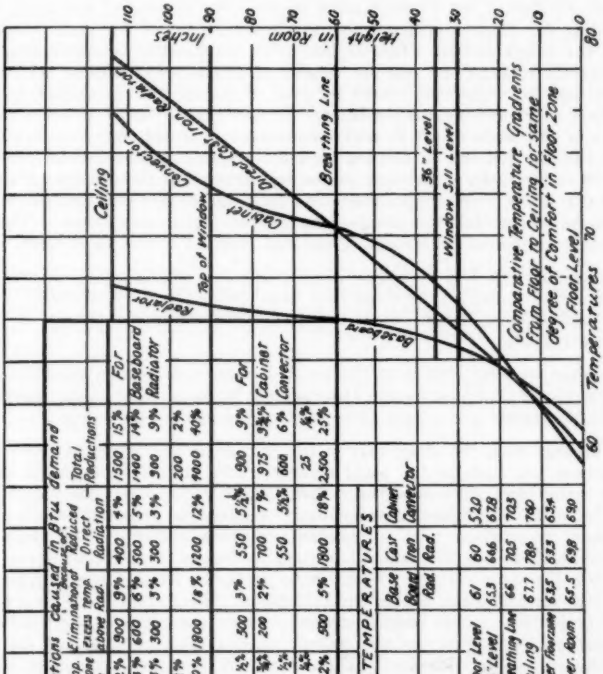

FIG. B. HEATING EFFECT DETERMINATIONS IN A TYPICAL ROOM REQUIRING 10,000 BTU OF HEAT DEMAND IN ZERO WEATHER AND 15-MILE WIND

sion implies a higher or lower heating effect so that it is natural that a high heating effect should be designated as, say 40 per cent, while a low heating effect might be 10 per cent. Since a 38-in., 5-tube cast-iron radiator may be regarded as unity then a radiator of 10 per cent heating effect uses 10 per cent less heat input than the cast-iron radiator, while a radiator of high heating effect may have 40 per cent heating effect and use 40 per cent less heat input.

2. In view of the foregoing, the saving in heat input means the same percentage in fuel saving, which when floor zone comfort is taken as the index, is the sought for result of heating effect. Thus a radiator of no heating effect percentage gives no fuel saving but a radiator of 40 per cent heating effect will give 40 per cent saving.

For these reasons the percentage method of designation as outlined previously appears to be the more logical and more readily understood by the lay mind.

A further criticism of the Trane-Scanlan paper is the plotting of the chart, Fig. 8. In the plotting for the convection heater the line drawn does not pass through the average of the points plotted. This results in an erroneously favorable comparison for the convection heater and further implies a relationship between wall surface temperature and Btu input that cannot be true. From the line as plotted, using the Trane method of expressing the coefficient, we find this to be 69 F when the wall temperature is 58, while at 66 F wall temperature it is still 69. A more correct location of the line and one in keeping with the findings of others gives coefficients of 73 and 78 respectively. Obviously as the temperature of the exterior of the wall approaches that of the room the heating effect ratio approaches zero and the cast-iron radiator would become fully as effective as the convection type heater. Therefore, the comparison as plotted by Messrs. Trane and Scanlan cannot be correct.

For a practical method of indicating heating effect in numerical values the writer worked out a plan shown on the chart here presented. The method admits of ready calculation and removes much of the mystery with which the theory of heating effect is now regarded.

It is not clear whether Professor Rowley had some such plan in mind in offering his paper on Essential Elements for Determining Heating Plant Requirements. At any rate his arguments give plenty of support to the method proposed.

It is to be seen from the chart that in the calculation for heating effect the heat loss calculations are made in the usual manner. This demand is taken as that imposed upon a standard cast-iron radiator, then for the radiator or heater whose heating effect is to be determined, a temperature gradient is prepared from test data gathered from comparison with a standard cast-iron radiator. The temperature gradients are laid out on the same chart and so located that temperatures in the floor zone are the same average. From these gradients, temperatures can be determined for wall surface, glass, ceiling and infiltration. Since from a cast-iron radiator the largest proportion of heat emitted streams directly upward over the window surface, the wall above, and across the ceiling, account must be taken of the higher surface temperatures over this area. These items are separately calculated and then compared with the known heat demand difference between the radiator in question and the standard radiator. The sum of the separate items is found to be less than the difference between the two radiators and this balance can be attributed to direct radiation. Thus the effect of direct radiation is determined by difference rather than by calculation. There seems to be no method at present known by which the effect of direct radiation can be exactly determined. This is a problem now under investigation in the writer's laboratory.

With the advance that has been made in the study of heating effect in the past few months there is good reason to expect that very shortly all types of space heaters will be graded according to their heating effect. The baseboard type of radiation shown on the chart has been under observation in the writer's laboratory for the past four years and is now put out on a strictly heating effect rating by evaluating the rating according to the plan here shown. As a result of this method of presentation all mystery has been removed and the explanation of its heating effect accepted as entirely logical, although it has been found possible as can be seen from the chart

to give it a heating effect rating as high as 40 per cent—a value that is truly staggering, when it is realized that this percentage also represents the actual saving in fuel over that obtained with a standard cast-iron radiator when the radiators are operating under zero weather conditions.

H. L. STILES (WRITTEN): The authors' conclusions verify certain experiments we have made relative to electric air heating radiation. Although we have not had the advantage of conducting our tests in a room in which the temperature conditions could be so closely controlled, the test results indicate a trend towards those secured by Trane and Scanlan.

The test room in which we worked measured 15 x 12 x 9 ft, with two 3- x 5-ft windows in an east wall. The door, 2½ x 7 ft, was located in the north wall. The wall and ceiling were insulated with approximately 4 in. of rock wool, while the floor was built up of 6 in. of concrete, sleepers, tar paper and flooring.

Preliminary to our tests we were of the opinion that the radiant type of electric heater was suitable, where an occasional, localized heat is required, while the purely convection type heater was intended primarily for continuous heating where the air was to be maintained at some predetermined temperature. There is, however, a deficiency of literature on this subject, so, therefore, we made our own investigation in this field.

The radiant type of heater used in our tests operates with electric coils at a bright red temperature and from previous radiometer tests showed a radiation efficiency of 37 per cent. It seemed apparent that this type of heater would be glaringly inefficient, since 37 per cent of its heat would be distributed over cold walls, doors, and windows to be lost by transmission.

The convection heater, on the other hand, since it showed 97 per cent convection efficiency, seemed to be the efficient heater for continuous heating since the radiation losses are absent and the air is continuously circulated and recirculated by natural draft through the *convector* and heated. Also the units, as contrasted with the radiant heater, operated at a *black heat*.

Table A gives some of the data from the convection and radiant heater tests, as well as the results obtained when a 14-in. fan was used to circulate the air in the room while the convection heater was tested.

TABLE A

Type	Watt-hr Used per Cu Ft per Deg. Rise (Raising Temp.)	Temp. Grad. Floor to Ceiling	Watts Input per Cu Ft per Deg. Diff. Between Inside and Outside Maintaining Temp.
Convection heater	0.074	18½	0.020
Convection heater with fan	0.066	7	0.036
Radiant heater	0.1203	11.5	0.019

The watt hours used per cubic foot per degree rise was the average value secured by dividing the total watt hour input to raise the temperature to 70 F at the 5-ft level, by the temperature rise and the cubical volume of the room. The temperature gradient was taken when the thermometer at the 5-ft level was 70 F. The watts input per cubic foot per degree difference was also an average value obtained by dividing the total watt hours consumed during the period of the test by time, temperature difference between inside and outside, and the cubical volume of the room. The outside temperature was in all tests between 30 and 40 F. Both heaters were rated at 3,000 w each.

The first test, on the convection heater, might be taken as a standard for comparison. The temperature gradient between floor and ceiling for this heater was 18.5 F, which is about average for a 9-ft ceiling.

It will be noted that by circulating the air in the room, the watt hours used per cubic foot, etc., were decreased as shown in test No. 2 when compared with test No. 1. This would indicate that, since the air was continually stirred up, the heat was not allowed to pocket at the ceiling, but was utilized in the living zone. Further, proof of this was evident from the decrease in temperature gradient from 18.5 to 7 F. On the other hand, the watts input, etc., was increased materially, indicating that the air movement over walls, ceiling and floor tends to increase losses to these areas.

The radiant heater in test No. 3 shows that during the heating up period the watt hours consumed was greater, due no doubt to the absorption of the radiant heat by the walls, ceiling, furniture, etc. This absorbed heat is to a large extent recovered later, although there is considerable wall transmission loss. No attempt was made to determine this loss quantitatively but it is possible that, though it may not be excessive, it is appreciable as later discussed.

It will be also noted that the radiant heater watts input per cubic foot per degree difference between outside and inside temperature, which was maintained at 70 F at the 5-ft level, was slightly lower than for the convection heater. This condition seemed improbable, but repeated tests gave verifying data. When it is considered that this radiant heater had 37 per cent radiation efficiency, these results are of significant importance. Our analysis of the data is that the heat radiation from the radiant heater falls on all exposed surfaces in the room, heat-soaking them until a condition of equilibrium is reached, when they in turn become effective as radiators and convectors. This extremely large area of low temperature heating area induces but a small temperature rise in the surrounding air. This is an ideal condition for heating since it results in a low temperature gradient between floor and ceiling.

Thus it seems that the radiant heat received and absorbed by the walls, etc., is partially lost to the outside, while the remaining portion is given up to heat the air in the room. The part given back into the room is utilized sufficiently efficiently as to more than compensate for the loss and results, therefore, in a condition better, or at least equivalent to the convection heater.

E. J. VERMERE (WRITTEN): Without question this paper is an important contribution to the progress of heating and in the future years will be considered as a milestone marking a definite step toward better engineering. For several years we have discussed heating effect, although we have not always used this term and many times where a fuel saving may have occurred due to heating effect, credit for this economy has been given other equipment in the system and not to the type of radiator used. This is the first example we have had where heating effect has been discussed beyond the theory stage and where a definite conclusion has even been attempted. If subsequent experiments bear out the conclusions reached in this paper, it will certainly prove that much of our highly advertised engineering ability is still based on thumb rules.

Most engineers will admit that in commercial practice it is impossible to obtain heat loss calculations within 25 per cent of actual requirements and that such a margin of error is certainly a heavy tax on the man who pays for the installation.

I would like the authors to give more of the information which led them to make the two following statements, *first:* "No material difference could be determined between heating effect coefficients for high and low radiators. If the results of the tests for the cast-iron radiators as given in Table 1 are plotted separately instead of being given as the average reults, it will be found that the 5-tube 26 differs materially from the 5-tube 38 and the wall radiator. A wider range of similar tests would answer this question." *Secondly:* The authors state, "likewise, no appreciable difference could be detected between a convection heater with a high stack and one with a low stack." It seems reasonable to believe that this would be true for

various heights of convection heaters of the same type where the outlet grille is below the breathing level, however, the question arises whether tests have been made with high outlets, especially where the outlet is near the ceiling line. If such tests have been made then our theory of air stratification which prevents proper heat diffusion with a high outlet must be incorrect. Actual experience with installations where outlet grilles are at or near the ceiling have indicated improper heat distribution and therefore the same heating effect factor could not apply.

Another basic theory of heat diffusion which we have accepted without much supporting data is ignored in these conclusions; that is the relation between heat diffusion and temperature of the air leaving the outlet. For all practical purposes, the unit heater is to be considered a convection heater. We also know that high outlet temperature units will not give as good diffusion as low outlet temperature units of equal capacity. On page 135 of THE GUIDE 1931 it is recommended that the delivery temperature generally should not be more than 70 F above room temperature desired. Therefore it seems reasonable to believe that outlet temperature will have a decided bearing on the heating effect factor.

The experience of the last 10 years in the application of unit heaters is conclusive evidence that the factor of heating is primarily a relation of convected heat. Many manufacturers of this type of equipment claim fuel savings of at least 25 per cent and in most cases this economy can only be attributed to the heating effect factors due to convected heat, thus supporting the conclusions of the authors.

This entire subject is of such great importance to everyone connected with the heating profession that I feel it would be in order for the Society to take immediate steps to either approve or disprove these claims by ordering our director of the research laboratory to begin at once the necessary experimental work incident to a thorough analysis of this subject. It is easy to forecast the controversies that will arise on this subject between advocates of different types of space heaters and our Society is the one agency which can analyze this subject in an unbiased manner and whose findings will be accepted by the profession.

KONRAD MEIER: These experiments were intended to show, why convection heaters require less heat input to maintain the same temperature within a room than exposed cast-iron radiators and whether the heating effect should be combined with rating.

For these tests the so-called cold room method was applied with the idea of creating similar and variable conditions for heat emission and distribution as are met in actual practice, excepting only the infiltration, which was disregarded for purposes of comparison. Unfortunately the apparatus tested is not described as might be expected. No indication at all is given regarding the type insulation and arrangement of the convection heaters, that would naturally bear on the result. Only on the exposed heating surface is the information sufficient to show the conditions under which it must have been operating.

In practice, radiators are installed in front of masonry. This assumes a higher surface temperature and therefore favors the useful circulation of air back of the unit and at the same time the radiant emission is reduced where not wanted. The sheet metal of the cold room on the contrary was kept cool by irrigation, while the temperature of the radiator, judging by the data on air leaving the same, ranging between 145 and 180 F, was much higher than usual for low-pressure steam or water heating and was exceedingly variable. Thus the radiant heat emitted toward the wall was not only increased, but also largely wasted. A comparatively slight source of losses with radiator heating, probably no smaller for built-in heaters, had been multiplied by the cold room test. Moreover, the transmission losses must have been increased by the current of superheated air rising from the radiators and coming into direct contact with the cooled sheet metal wall, whereas, ordinarily the location of the radiator underneath the window does not mean increased losses through the glass, because its inner surface is colder than the irrigated sheet metal and creates

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a down draught. With the heating surface properly spread below the usual window sill, this air is mixed and deflected toward the room. The infiltration, if anything, helps to obtain that effect. Evidently, the conditions for testing the cast-iron radiators were far from normal and decidedly adverse in regard to emission as well as the utilization of the heat.

These conditions also explain why the wall radiator did not make a better showing. The waste of heat must have been still greater due to: (1) the spread of surface in front of the cold sheet metal and, (2) installation near the wall, resulting in excessive radiation and convection, of which the greater part was lost. These tests are, therefore, still further from proving any superiority of the convection heaters. However, they do confirm that flat radiation should not be set too close, so that the broad surface facing the wall is utilized, as it should be, for heating liberal volumes of air to moderate temperature, which reduces the tendency to overheat the upper zone. The tests also show that thin walls behind such radiators should be insulated, just as convection heaters in outside walls need extra protection.

In view of the large and variable differences of temperature between floor and ceiling it is evident, that neither the average nor the readings at any particular point could have been a safe standard for comparison. There are too many chances of error, even without the uncertain factor of infiltration and the unknown influence of the walls not irrigated. In any event, since the latest types of convection heaters probably were used in these tests, the question may well be raised, whether comparisons ought not to be made with exposed radiation as it should be, rather than with indifferent practice, even though the latter is still common. It has been well established by this time, that radiation under windows produces nearly uniform temperatures and best room heating effect, but only if the surface is spread out for about the width of the glass, so as to meet indraught and downdraught. Furthermore, the best results are not attainable, unless the temperature of the heat carrier is regulated to suit the requirement, so as to assure continuous and appropriate effect rather than a service of varying intensity. With the former it would not be necessary to resort to different sizes of radiators for testing, which leads to inaccuracies, nor to abnormal steam temperatures. With rational regulation the gradient would be better and the readings more reliable.

The authors have attempted to give some further proof of their contention by the example of a hot air heating plant, for which the input of a zero-day seemed to correspond fairly with the heating effect calculated from their tests. No records are given of inside and outside temperatures. However, even these could not be accepted as a confirmation, unless extended over a longer period, since the input of a single day bears a most uncertain relation to the calculated transmission. It is not only influenced by sunshine and the ever varying infiltration, but also governed largely by the thermal state of the building or the weather of previous days.

Additional tests were made for providing the existence of radiant energy losses with plain cast-iron and nickel-plated sections in hot and cold testing rooms. The description of the apparatus and equipment is also inadequate. It is stated, that with the cold room test the same breathing line air and wall temperatures were maintained as previously, but it is not made clear if this refers also to the hot room test. That the breathing line is not the proper basis for comparing comfort, is now sufficiently established. Better conditions are obtained at lower dry-bulb temperature within a lower zone. No data on the gradient are given for these tests, but it may be assumed that in the cold room the radiant heat was again carried off by the spraying water, while the hot room observations were altogether too short for safe conclusions. If these had been correct and fully 50 per cent of the heat could be saved by nickel-plated radiation, we would do better to use these in place of convection heaters. However, the tests merely confirm that the heat emission of polished metallic surfaces is much smaller than that of plain or painted cast iron. The

comparative utilization is not demonstrated. No attempt whatever was made to account for the radiant heat and to show its effect on the surroundings and on comfort. On the contrary, any such rise would be prevented in the cold room, which shows the fallacy of this method of testing for the present purpose. It is characteristic, that the investigation should be made with wall temperatures up to 66 F, a temperature which may often be observed outside the heating season, when no heat input at all is needed because of the perfect gradient and a natural state of the relative humidity. If the walls had been allowed to cool off until heat would be necessary, mild radiation would be effective even before the temperature of the surroundings and of the air could rise. But even in the hot room, where an increase in the interior wall temperature might be observable, it would not mean a corresponding increase of losses, since most of the radiant heat is generally absorbed by the inside walls, floors and the contents of the room and therefore contributing to comfort.

The comfort zone, as experimentally established for judging ventilation, proves sufficiently that the dry-bulb temperature is only one of several factors involved and that wholesome conditions are attainable at lower temperatures than generally assumed. A similar chart worked out for heating, with due regard for the thermal state of the surroundings and position radiant effect, might serve as a basis for comparison. It would show how far the dry-bulb temperature could safely be lowered at such humidity as would naturally result, thereby approaching pleasant outdoor conditions. We should not only avoid waste in the application of the heat, but we should also reduce the requirement. We should produce heating apparatus which will give comfort within the occupied space at lowest air temperature, which means proper regulation and best distribution of the heat, favoring the lower zone. To this end the radiant effect is generally useful, because it will raise the sensible temperature without warming the air. If the effect of a system on persons and the cost of the desired results would be thoroughly investigated, it might be shown that the 70 or 72 F now demanded, in reality include a factor of safety for imperfect heating. The fact is, that where different methods prevail, the standard living room temperature is lower. About 64 F is widely accepted in Central Europe.

As the inquiry is broadened, it becomes apparent also that room heating efficiency should not be expressed in percentages, because it is impracticable to determine the relation of the calculated heat requirement to the actual input with any degree of assurance. But the heating of a room can always be judged by the temperature gradient, the freedom of air currents and irritating dust, and the relative humidity, all of which contribute to comfort at moderate dry-bulb temperature and make for economy. And it should be clear from all the recent experimental data, that the heating effect is not altogether due to design, but also is dependent largely on selection and disposition of the heating surface. Thus the tubular patterns will be more effective underneath taller windows requiring a concentration of surface, whereas the wall type would be more advantageous for ordinary heights. Many similar and self-evident facts should at once exclude the idea of combining the room heating effect with rating. On this point the conclusion of the authors is rational. But instead of basing the rating on the output at the grille, it would seem more logical to adhere to the older way of input or condensation, which alone would lead to proper calculation of piping and boilers. And finally, the performance should always be tested with the heaters in location for which they are intended and designed. It is the task of the engineer to select and apply them according to the need of radiant or air heating capacity in a given situation.

MR. TRANE: May I call attention to one thing: that we have proved at least that in a tin building in a rainstorm we have a 30 per cent reduction of heat loss. Also I think we have proved that in a gale a tin building will at least save us that much in fuel consumption. It might be assumed that in a greenhouse under similar con-

ditions results would be the same. Probably in sun porches the results would be identical.

We are only trying to find the truth and any way in which the Society can guide or help us in the matter of getting the correct answer to this problem will be greatly appreciated.

PRESIDENT CARRIER: This has been a very interesting paper and a very interesting discussion of a subject that is of great importance.

No. 907

STUDY OF PERFORMANCE CHARACTERISTICS OF OIL BURNERS AND LOW-PRESSURE HEATING BOILERS

By L. E. SEELEY[1] AND E. J. TAVANLAR,[2] NEW HAVEN, CONN.
MEMBERS

This paper is the result of research conducted at Yale University, New Haven, Conn., in cooperation with the A. S. H. V. E. Research Laboratory and the American Oil Burner Association.

THE object of this study was to determine the performance characteristics of oil burners and low-pressure heating boilers when operating under various conditions. This report includes the results of numerous tests made with several oil burners and low-pressure heating boilers.

APPARATUS USED IN THE STUDY

The apparatus tested to date comprises four oil burners and four low-pressure heating boilers which are designated by letters *OB* and *B* respectively. For example, *OB 1—B 2* means the combination of oil burner No. 1 and boiler No. 2. Fig. 1 is a photograph of the set-up. A brief description of the boilers tested is given in Table 1.

TEST SET-UP AND PROCEDURE

The test set-up and procedure was in substantial agreement with the A. S. H. V. E. Code for Testing Steam Heating Boilers Burning Oil. See TRANSACTIONS of the Society, Volume 37, 1931.

Fig. 2 shows a diagram of the boilers. In addition to the usual equipment it should be mentioned that each boiler was equipped with an automatic draft

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regulator. An audiometer was used for sound measurements. The instrument is calibrated in terms of sensation-units, a standardized unit of sound. By means of a special ear piece the sound from the burner and from the audiometer enters the ear and the sound from the burner is measured by comparison with the audiometer sound.

TYPES OF TESTS RUN AND LEADING RESULTS

In order to obtain results which could be compared it was important to plan and carry out all tests according to a definite procedure. Tests were run under a variety of conditions necessary to determine performance character-

FIG. 1. GENERAL VIEW OF SET-UP OF APPARATUS

istics. These were designated by a series letter and the following presentation and discussion of leading results is included to illustrate the purpose of the various tests.

Series A. These were continuous tests with No. 1 oil (*i.e.,* fuel burned continuously at a uniform rate throughout test). Certain operating conditions were fixed in order to be reasonably sure that the burner was performing in essentially the same manner in each boiler. The CO_2 was standardized at 10

TABLE 1. DESCRIPTION OF LOW PRESSURE HEATING BOILERS TESTED*

	Designation of Boiler			
	B1	*B2*	*B3*	*B4*
Area of base, square feet	5.04	3.28	6.25	3.18
Water capacity to water line, lb	345.5	303.5	236.3	747.0
Heating surface, square feet	91	52.6	58.2	77
Total furnace volume, cubic feet	11.83	8.08	18.48	10.48
Average furnace width, inches	22.0	22.0	30.0	21.0
Average furnace length, inches	33.0	21.5	30.0	28.7
Average furnace height, inches	28.2	29.5	35.5	30.0

* These boilers were requested for an output of 1000 sq ft of "radiation" and represented the nearest approach to this rating made by the manufacturer in the spring of 1929.

per cent, and with all burners except *OB 5* the minimum fuel burning rate was the lowest rate that could show 10 per cent ($\pm$ 0.4) CO_2 without apparent smoke.

The furnace draft was standardized at 0.02 in. of water and another test was run to determine the maximum rate of fuel burning possible with 10 per cent CO_2 and the draft of 0.02 in. Rates of oil burning exceeding this maximum

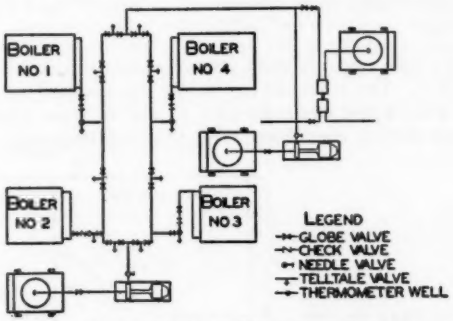

FIG. 2. DIAGRAM OF FEEDWATER PIPING

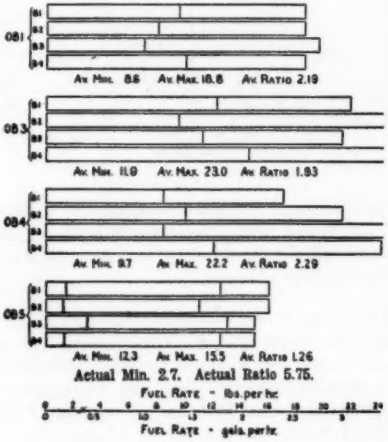
OB1
Av. Min. 8.6 Av. Max. 18.8 Av. Ratio 2.19
OB3
Av. Min. 11.9 Av. Max. 23.0 Av. Ratio 1.93
OB4
Av. Min. 8.7 Av. Max. 22.2 Av. Ratio 2.29
OB5
Av. Min. 12.3 Av. Max. 15.5 Av. Ratio 1.26
Actual Min. 2.7. Actual Ratio 5.75.
Fuel Rate - lbs. per hr.
Fuel Rate - gals. per hr.

FIG. 3. MAXIMUM AND MINIMUM FUEL RATES OBTAINED WITH 10 PER CENT CO_2 AND 0.02 IN. WATER FURNACE DRAFT LIMITS

were generally run if the rate of oil feed was capable of an increase. In this last test the furnace draft exceeded 0.02 in. water.

1. Fig. 3 shows maximum and minimum oil burning rates obtained within the 10 per cent CO_2 and 0.02 in. draft limitations imposed. It should be understood that a study of this type requires that some CO_2 percentage and furnace draft be chosen in order to be able to make valid comparisons. It should also

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be pointed out that 10 per cent CO_2 is a reasonable working standard. The results reported in this paper indicate the performance that could logically be expected under the usual operating conditions. The furnace draft of 0.02 in. of water may be considered low but it seems desirable as will be shown. Furthermore, it is obvious that there will be maximum and minimum fuel burning rates for whatever percentage of CO_2 and furnace draft is fixed upon. Both values will shift together in the same direction, however, so that the range or difference between maximum and minimum rates may stay much as shown by Fig. 3.

OB 5 is shown with two minimum values because this is a continuous operating type of burner. The attempt to go below the standard minimum conditions of 10 per cent CO_2 would cause the CO_2 of any of these burners to diminish. *OB 5* represents such a condition. The point of importance in this case is

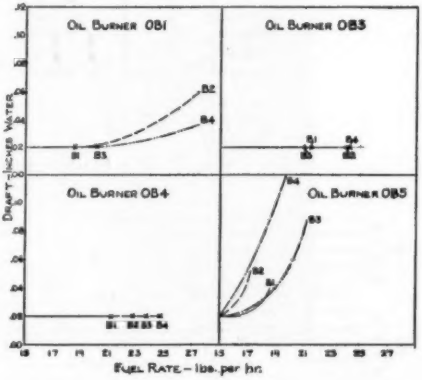

FIG. 4. GRAPHS SHOWING THE RELATION OF FUEL BURNING RATE AND FURNACE DRAFT FOR VARIOUS OIL BURNERS

that an operating fuel rate ratio would have to be greater than the average value of about 2.0 shown for *OB 1, OB 3,* and *OB 4,* because, due to its continuous operation, this type must possess sufficient flexibility to meet exactly all fluctuations in heating loads caused by weather changes.

2. Fig. 4 shows another point in relation to maximum fuel burning rates and furnace draft. It will be noted in the cases of *OB 1* and *OB 5* that oil burning rates in excess of 20 and 15 lb per hour respectively, required the furnace draft to be increased. This merely indicates that the increased air required had to be assisted into the furnace by an increase of furnace draft. *OB 3* and *OB 4* did not reach such a limit during these tests.

All oil burners would probably show curves like those of *OB 1* and *OB 5* if forced sufficiently unless the burner was limited by its ability to supply oil. *OB 3* was so limited in this case, but not *OB 4.* Practically, this curve suggests that oil burner manufacturers' ratings should be made at a point somewhere near the 0.02 in. of water furnace draft value. The fuel rates at maximum might be somewhat increased with no change in air, thus producing CO_2 greater than 10 per cent, but this increase would be comparatively small at best.

In addition to the maximum and minimum Series A tests just described, a number of similar tests were also run at intermediate outputs, so that performance curves might be obtained. The complete results obtained from Series A tests are included in Figs. 29 to 44. Values obtained are overall efficiency, flue temperature, fuel rate, furnace draft, smoke hood draft and per cent CO_2. Other values have been obtained from the test data which will be enumerated.

3. Fig. 5 shows as the first fact that the boiler efficiency curve (*i.e.*, its shape) preserves its general character regardless of the oil burner. One would conclude from this evidence that the shape of an efficiency curve is largely a characteristic of the boiler. The fact that all the curves for one boiler do not

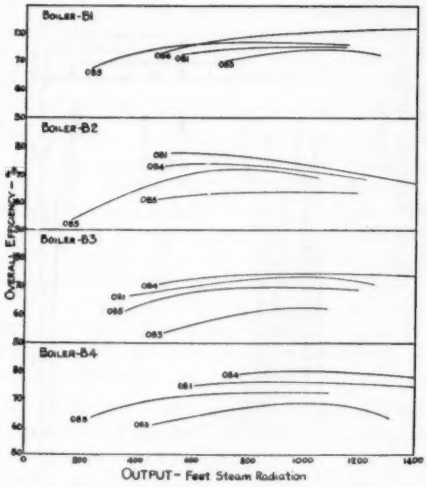

FIG. 5. OVERALL EFFICIENCY CURVES, GROUPED ACCORDING TO BOILERS

coincide indicate that the oil burner does influence the value of the curve but not its character.

4. Fig. 6 (*a* and *b*) shows the maximum efficiencies obtained irrespective of boiler output at which said efficiency occurred. It is interesting to note that the range is from 62.3 per cent to 81.4 per cent for all boilers. That various oil burners produce different maximum values in the same boiler is also apparent, *OB 3* being the lowest in every case and *OB 4* being highest in three instances. The boilers in order of highest efficiency may be listed as *B 1*, *B 4*, *B 2* and *B 3*.

5. Fig. 7 shows the boiler outputs at which the maximum efficiencies in Fig 6 were obtained. Maximum efficiencies for *B 1* occurred in all but one case at higher outputs than the other boilers. *B 3* and *B 4* show the same general results and *B 2* gave lowest outputs at maximum efficiency. Referring to Figs. 6 and 7, *OB 4—B 2* shows 74.4 per cent efficiency at 675 sq ft of equivalent direct "radiation" output and *OB 4—B 1* 81.4 per cent at about 1400 sq ft of "radiation" output. This illustrates what may be an important factor in

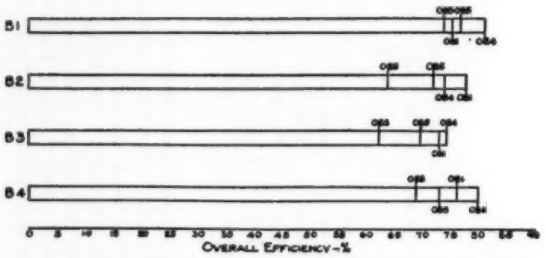

FIG. 6-*a*. MAXIMUM OVERALL EFFICIENCIES IRRESPECTIVE OF BOILER OUTPUT RATE, GROUPED ACCORDING TO BOILERS

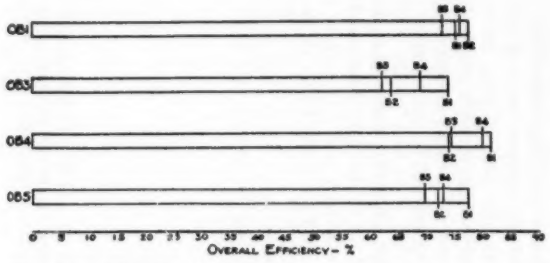

FIG. 6-*b* (above). MAXIMUM OVERALL EFFICIENCIES IRRESPECTIVE OF BOILER OUTPUT RATE, GROUPED ACCORDING TO BURNERS

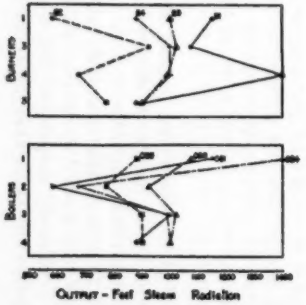

FIG. 7. CURVE SHOWING OUTPUTS AT WHICH MAXIMUM EFFICIENCIES OCCURRED, GROUPED ACCORDING TO OIL BURNERS AND TO BOILERS

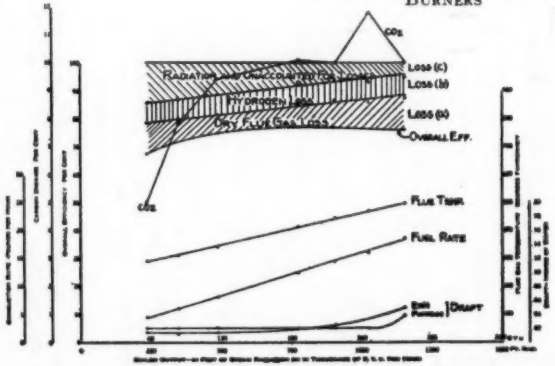

FIG. 8. PERFORMANCE CHART *OB 5—B1* HEAT BALANCE

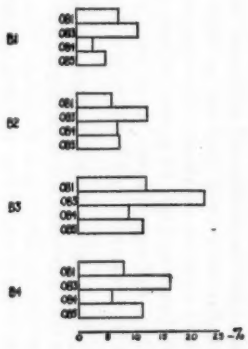

FIG. 9. PER CENT OF RADIATION AND UNACCOUNTED FOR LOSS (*c*) AT 17.5 LB-HOUR FUEL RATE, GROUPED ACCORDING TO BOILERS

practice, namely, that in considering the application of two boilers, maximum efficiency should be stated in connection with output at which this efficiency is obtained.

6. Fig. 8 shows graphically an example of a complete heat balance. The losses are due principally to (*a*) loss in dry flue gas; (*b*) hydrogen or water vapor loss, and (*c*), radiation and unaccounted for losses. Loss (*a*) depends upon the weight of dry flue gas formed per pound of oil and the temperature of the flue gas. Loss (*b*) depends upon the hydrogen content of the fuel and the flue temperature. Loss (*c*) is found by difference in order to account for 100 per cent of the heat produced by the fuel.

It should be noted that loss (*a*) generally increases with boiler output because flue temperature increases also. Loss (*b*) changes very little, increasing only slightly with increased boiler output. Loss (*c*) decreases with output because radiation losses remain substantially constant at all outputs, hence causing an actual percentage decrease as the boiler output increases.

An especially interesting fact about loss (*a*) in the case shown in Fig. 8 is that it is greater at 250 sq ft of "radiation" output than at 500 sq ft of "radiation," even though the flue temperature is lower at 250 sq ft of "radiation." The explanation is seen in the CO_2 curve which starts falling rapidly from about 500 sq ft of "radiation." A lower CO_2 means greater excess air and an increased weight of dry flue gas per pound of oil. Even though the gas temperature is lower at 250 sq ft of "radiation," the increase in weight of dry flue gas more than offsets it. Losses (*b*) and (*c*) are not greatly changed by excess air but big differences may occur in loss (*a*). This is the reason why oil and air adjustments should be measured by an Orsat to be sure that there is no unreasonable excess of air.

7. A further study of loss (*c*) Fig. 8 should furnish some clue as to why the same boiler may show changes in efficiency with various burners operating under standard series *A* conditions.

Fig. 9 shows these losses grouped according to boilers and based on identical fuel burning rate (*i.e.*, 17.5 lb per hour). A rate of 17.5 lb per hour was chosen as a reasonable one, giving an average boiler output of about 1,000 sq ft of "radiation." The most obvious fact is that *B 3* shows higher average losses than the other boilers. This can be accounted for by the fact that *B 3* has a high base, consequently a large exposed area, which is not water backed and as a result may allow higher radiation losses than the other boilers.

Fig. 10 shows the same values grouped according to oil burners. This shows that *OB 3* apparently causes higher radiation losses than the other burners. This can be accounted for by the fact that up to the present time the results obtained with this burner indicate that it heats the floor and ashpit sidewalls more than the other types and therefore causes higher radiation losses. It is evident that boiler and oil burner both may influence this value when all variations in design are considered.

8. Fig. 11 shows flue temperatures grouped according to boilers all at a fuel rate of 17.5 lb per hour. The temperatures vary from 380 to 720 F and it is obvious that temperatures for *B 2* average higher than the other boilers. Aside from this fact there is nothing else shown of a conclusive nature.

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9. Fig. 12 shows a boiler characteristic which is the draft loss through the boiler for a fuel burning rate of 17.5 lb per hour. The values for one boiler and various oil burners are not the same in spite of the fact that the weight of flue gas per pound of oil is the same in all cases. It is apparent that draft losses vary from 0.005 to 0.05 in. of water, which is a wide variation. That the

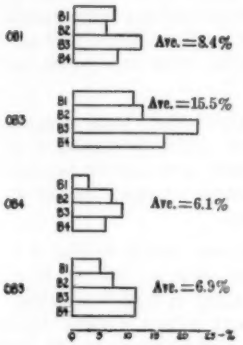

FIG. 10. PER CENT OF RADIATION AND UNACCOUNTED FOR LOSS (*c*) AT FUEL RATE OF 17.5 LB PER HOUR, GROUPED ACCORDING TO OIL BURNERS

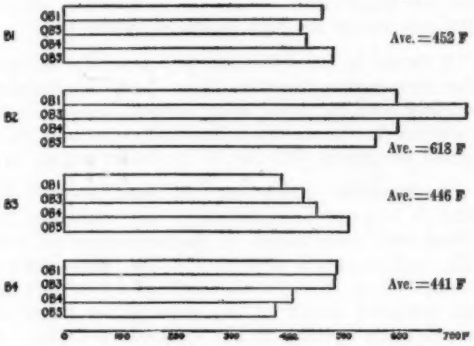

FIG. 11. FLUE TEMPERATURES AT FUEL RATE OF 17.5 LB PER HOUR, GROUPED ACCORDING TO BOILERS

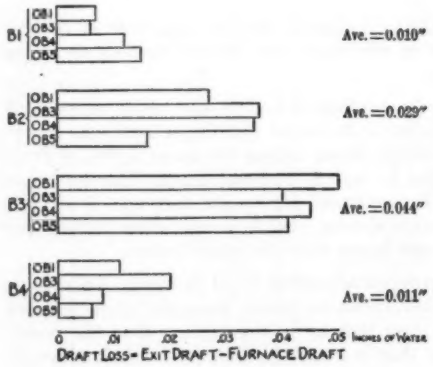

FIG. 12. DRAFT LOSS THROUGH BOILERS AT A FUEL RATE OF 17.5 LB PER HOUR

boilers may be classified from high to low as to amount of draft required is evident as follows: *B 3, B 2, B 4, B 1.* Boilers of this type operating on comparatively low chimneys should not indicate too high a draft loss but there is some evidence that the resistance need not be reduced to a minimum either.

10. Another feature of interest shown by series *A* tests is the average heat absorbed per square foot of boiler heating surface for fuel rates of 17.5 lb

per hour in all cases. Fig. 13 shows rates ranging from 2,700 to 4,700 Btu absorbed per square foot of heating surface per hour. The boiler showing the highest value is *B 2*, and in descending order, *B 3*, *B 4* and *B 1*. The average overall efficiency of all four burners in each boiler at the same fuel rate of 17.5

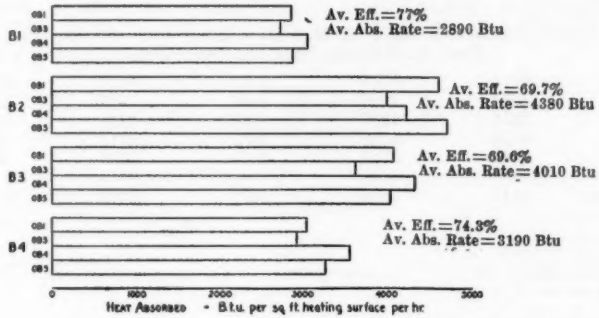

FIG. 13. AVERAGE BTU ABSORBED PER HOUR PER SQUARE FOOT OF BOILER HEATING SURFACE AT A FUEL RATE OF 17.5 LB PER HOUR

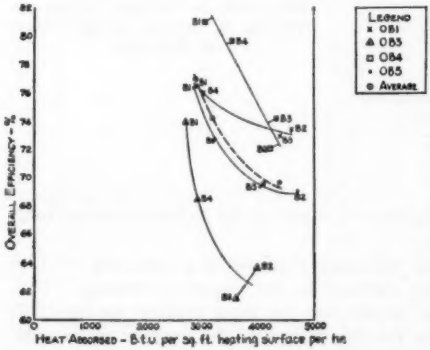

FIG. 14. EFFICIENCY AT FUEL RATE OF 17.5 LB PER HOUR VS. RATE OF HEAT ABSORPTION

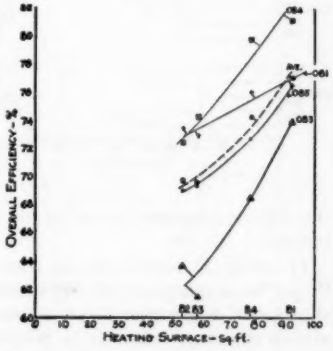

FIG. 15. EFFICIENCY AT FUEL RATE OF 17.5 LB PER HOUR VS. HEATING SURFACE

lb per hour is as follows: *B 2*—69.7 per cent, *B 3*—69.6 per cent, *B 4*—74.3 per cent and *B 1*—77.0 per cent.

Fig. 14 shows clearly that higher efficiencies are accompanied by lower heat absorption rates but that these values vary with the burner. It is evidently a characteristic of *OB 4* that it can obtain heat absorption rates at better efficiency than other types of burners. *OB 1* has similar characteristics at high heat transfer rates. Both *OB 1* and *OB 4* have a similar type of flame.

Table 2 shows the heat absorption rate per square foot of heating surface per hour at the point of maximum efficiency. The minimum rate is 2,329 Btu with *OB 5—B 1* and the maximum is 4,225 Btu with *OB 3—B 2*. It is interesting to note that the average rate of each of the burners in the different boilers is between 3,000 and 3,600 Btu. The average rate of each boiler with

TABLE 2. HEAT ABSORPTION RATE AT MAXIMUM EFFICIENCY IN BTU PER SQUARE FOOT OF HEATING SURFACE PER HOUR

	OB 1	*OB 3*	*OB 4*	*OB 5*	Average
B1	3032	2835	3692	2329	2972
B2	2674	4225	3080	3536	3379
B3	4124	4206	4082	3711	4031
B4	2743	3117	3117	2805	2945
Average	3143	3596	3493	3095	

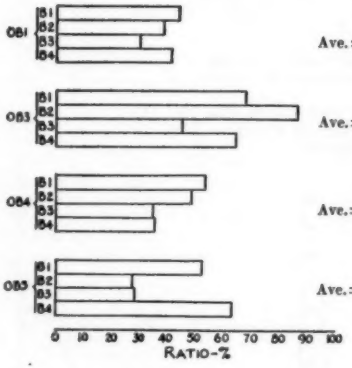

FIG. 16. RATIO OF FURNACE VOLUME *B* TO FURNACE VOLUME *A* (FURNACE VOLUME *A*, BEFORE INSTALLATION OF BURNER; FURNACE VOLUME *B*, AFTER INSTALLATION OF BURNER)

the different burners is not within as short a range as the average rate of each burner.

11. Fig. 15 represents the overall efficiency obtained at a fuel rate of 17.5 lb per hour compared to the heating surfaces in the respective boilers. This shows that for a given rate of heat generation the most heating surface will absorb the most heat. It is possible for the design of the surface to be such that two boilers of equal surface will not absorb an equal quantity of heat. The average difference between *B 2* and *B 3* is negligible but loss (*c*) Fig. 9 gives a clue in explanation thereof.

12. *Heat Liberation Rates in Furnace:* Various values have been submitted for the desirable rate of burning oil per cubic foot of furnace volume. First of all there are two furnace volumes—(*a*) the space available for fuel burning within the boiler as supplied by manufacturer—called furnace *A*—and (*b*) the available fuel burning space after the burner has been installed—called furnace *B*. It is evident that various types of burners will occupy different proportions of furnace *A* according to type.

Fig. 16 illustrates the per cent of furnace *A* left over after the burner installation, showing that furnace *B* may be from 27.5 to 86.5 per cent of furnace

A. This is a considerable variation. *OB 3* is shown as having a greater per cent of furnace *A* for combustion purposes; the others are about the same. This is caused by the nature of the furnace bricking and the proportion of burner actually placed within the furnace. The true measure of heat liberation rate in thousands of Btu per hour per cubic foot of furnace volume should naturally be based on the volume actually available which is furnace *B*. See A. S. H. V. E. Proposed Code for Testing Steam Heating Boilers Burning Oil Fuel for definition of furnaces *A* and *B*.

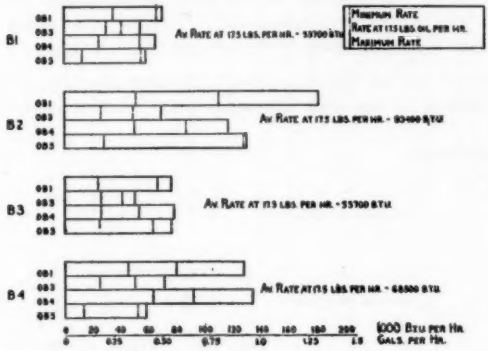

FIG. 17. MAXIMUM AND MINIMUM HEAT LIBERATION RATES IN THOUSANDS OF BTU PER CUBIC FOOT PER HOUR OF FURNACE *B*

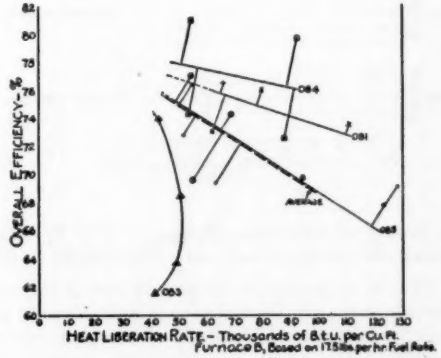

FIG. 18. EFFICIENCY AT FUEL RATE OF 17.5 LB PER HOUR VS. HEAT LIBERATION RATE

Fig. 17 shows minimum, 17.5 lb per hour and maximum rates per cubic foot of furnace *B*. The lowest rate shown is 12,500 and the highest rate 181,000 Btu per cubic foot of furnace *B* per hour. At 17.5 lb per hour the minimum is 40,800 and maximum is 127,500 Btu. *B 1* shows the lowest average rate at 17.5 lb per hour at 53,700 Btu and *B 2* the highest at 93,400 which is equivalent to 2.73 to 4.74 lb of oil per cubic foot of furnace *B* per hour. *OB 3* shows lowest average for reason shown by Fig. 16.

The general relationship between heat liberation rate and efficiencies is shown by Fig. 18. Both values are those obtained at a fuel rate of 17.5 lb per hour.

At a glance it appears that the curves on the figure are far off from the points. This impression will be corrected if the fact is noted that the vertical scale is magnified and that the maximum deviation from any of the curves is just a trifle beyond that allowed for experimental error. The significant facts about these curves are their negative slope to the right. Also, it should be noted that the curves for *OB 4, OB 1,* and *OB 5,* in this order, lie above the average curve and the curve for *OB 3* is below this curve.

The curve for *OB 3* has a form and direction different from the three other curves. The *OB 3* curve shows an increase in efficiency accompanied by a decrease in heat liberation rate up to a certain point. Beyond this point a decrease in heat liberation rate is accompanied by a decrease in efficiency.

In order to ascertain whether or not there is a desirable heat liberation rate per cubic foot of furnace volume *B*, the heat liberation rates were determined

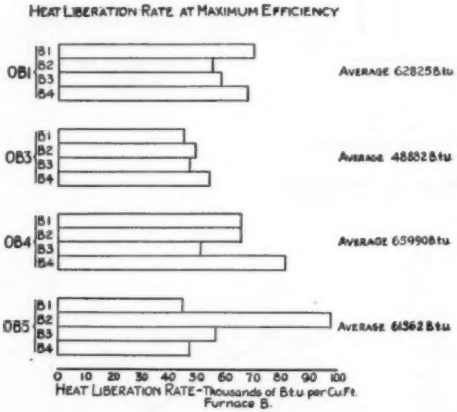

FIG. 19. HEAT LIBERATION RATE AT MAXIMUM EFFICIENCY

at points of maximum efficiency. Fig. 19 shows that heat liberation rates at maximum efficiency may vary from 44,600 to 97,400 Btu per hour.

Due to variations in furnace *B* and in burner design it is not clear that any specific value can be chosen as a desirable rate of fuel burning. Any rate would seem reasonable which produces complete combustion and gives an amount of heat that the boiler can efficiently absorb. Heat liberation rate cannot be divorced from the heat absorbing possibilities of the boiler.

13. *Sound measurements:* Figs. 20 to 23 show the sound emitted classified according to burners, measurements being made by the method previously described. Two measurements were made, one with and one without combustion to detect whether or not combustion sounds were present. *OB 1* and *OB 3* were found to be free from combustion sounds; *OB 4* showing an increase of 5 sensation units and *OB 5* of 10 sensation units. *OB 1* and *OB 5* gave the highest sound measurements. Average values indicate the following desirability of the burners as to sound: *OB 4, OB 3, OB 5, OB 1.*

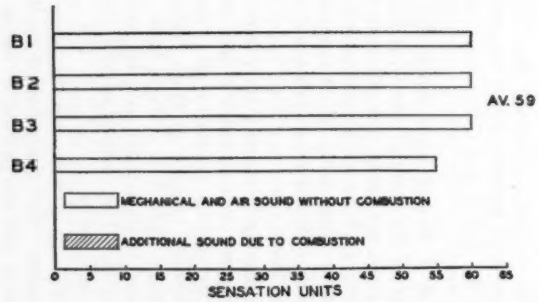

FIG. 20. SOUND EMITTED BY BURNER *OB 1*, SERIES *G*

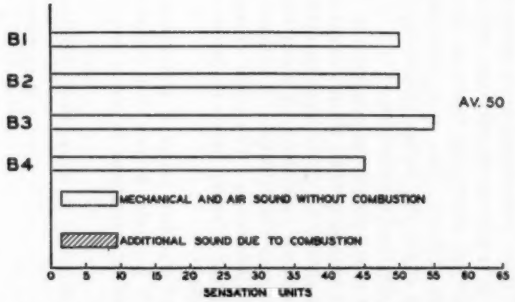

FIG. 21. SOUND EMITTED BY BURNER *OB 3*, SERIES *G*

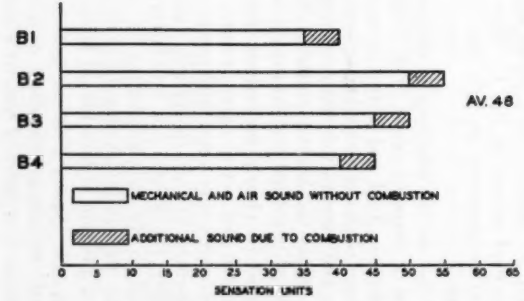

FIG. 22. SOUND EMITTED BY BURNER *OB 4*, SERIES *G*

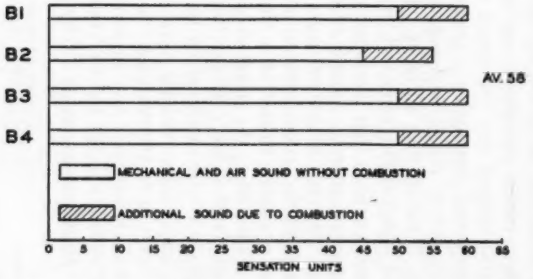

FIG. 23. SOUND EMITTED BY BURNER *OB 5*, SERIES *G*

The discussion of an item of this sort is a delicate matter because of variations in personal sensibilities and because of little background based on facts. It is also realized that the sound measured in a laboratory may not indicate what an actual installation may do on account of sound and vibration transmissions due to piping, building construction, chimney, etc. It is reasonably certain, however, that all other things being equal, the burner producing the smallest sound at the boiler will be generally the most satisfactory.

It seems evident that the sound impression is not strictly proportional to the measurements shown. Sounds of 40 and 50 sensation units do not seem unpleasant, whereas 55 and 60 seem quite the contrary. It is likewise evident that most of the sound is due to mechanical and air conditions. Fan air inlets may be profitably studied in a similar manner to the attempts of automotive engineers to reduce the sound of air entering carburetors. The effect of boiler types on sound is not very conclusive although *B 4* seems better in two cases. No definite reason can be given for this, however.

The sound measurements were made under standard series *A* conditions at about 1,000 sq ft of "radiation" output. It is generally known that variations in fuel-air proportions may in some cases produce variations in sound, but this special matter was not included in these tests.

14. *Power requirements:* The power required by various burners follows:

Burner	Power Required (Watts)	Burner	Power Required (Watts)
OB 1	58.2	OB 4	85.8
OB 3	181.4	OB 5	75.5

These are average values for all boilers at the different fuel rates at which series *A* was run.

Series B: Series *B* is a set of tests similar to series *A*, except that No. 3 oil was used instead of No. 1. This was done with burners which the manufacturer recommends for oils No. 1, 2 and 3. *OB 1* and *OB 3* were such burners.

15. So far as efficiency is concerned, *OB 1* showed small differences in amount or shape of curve. *OB 3* showed up on the average somewhat better with the No. 3 oil and in two cases the shape of the curve was different, the maximum efficiency occurring at lower outputs. In 7 out of 8 combinations of boilers and burners the efficiency was somewhat lower with No. 3 than with No. 1 oil at the highest outputs. The overall differences otherwise are not considered important and no graphs will be shown.

It must be clearly understood that the foregoing comments refer only to combustion and not to other features of operation, such as ignition and other factors which may show differences caused by various grades of oil. There are nowhere in this report any facts or opinions relative to those features necessary for safety and satisfactory automatic operation. These tests are confined to the heat-generating or oil-burning characteristics of the burner and the factors influencing these characteristics. The concern of this study is not a question of how the oil ignites but how it burns.

Series C: Series *C* is one test designed for one purpose and that purpose is merely to indicate the effect of a draft change. The conditions of the test are similar to a series *A* test, of a corresponding fuel rate, giving an output of

about 1000 sq ft of "radiation," with only one difference and that is the change in furnace draft from 0.02 to 0.07 in. of water. Does this affect combustion efficiency and flue temperature?

16. Fig. 24 shows the effect on combustion and efficiency for a 0.05-in. water change in furnace draft. *OB5, OB 4* and *OB 1* show a few cases where the draft difference was other than 0.05 in. and are so marked. The upper of each pair of bars is efficiency, that portion shaded showing the decrease caused by draft increase. The lower of each pair of bars represents the standard series *A* 10 per cent CO_2 value, the shaded portion showing the decrease in CO_2. The decrease in efficiency may be from zero to 17.7 per cent and CO_2 from 1 to 4.5 per cent, not including *OB 5* and those of *OB 4* and *OB 1* in the same

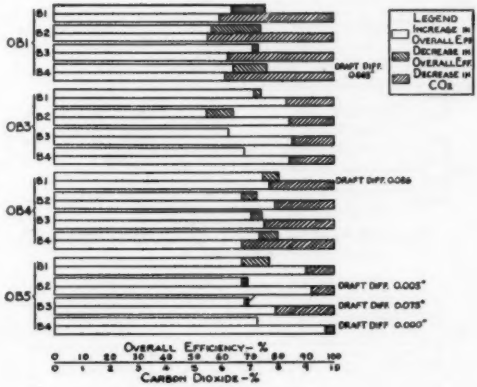

FIG. 24. COMPARISON OF EFFICIENCIES AND CO_2 OF SERIES *A* AND *C* AT THE SAME FUEL RATE. DRAFT DIFFERENCE 0.05 IN. WATER EXCEPT WHERE NOTED

category as *OB 5*, for the reason already stated. It should be noted that *OB 1* is particularly influenced by draft change, that *OB 4* is moderately influenced, and that *OB 3* and *OB 5* are somewhat variable.

17. Fig. 25 shows the effect on flue gas temperatures. The upper bar of each pair shows flue temperature, the shaded portion shows the increase in temperature due to increase in draft, and the unshaded shows the decrease. The results are not uniform, but the majority of cases indicate that flue temperatures may increase when the excess air increases. This aggravates the loss because not only is the weight of dry flue gas increased, but its temperature also—a double penalty, so to speak.

There are, of course, some cases where the reverse is indicated. *OB 1*, which showed the biggest drop in CO_2, shows the largest average flue temperature increase. It is not certain that the result (*i.e.*, whether flue temperature increases or decreases) can be foretold in all cases. It is thought that a boiler of small heating surface may show a decrease and vice versa an increase. Note that *B 1* shows an increase in all cases.

18. Series *C* shows obviously that furnace draft is one of the important factors in burner operation. It must be adjusted at the same time that the

oil and air are being set and it must be kept at a constant value just as oil and air should.

Series D: Series *D* is again a single test for a single purpose. All conditions of a series *A* test were obtained at an output of about 1000 sq ft of "radiation" with one exception. This exception is CO_2 and it is made as high as possible without visible smoke, according to the best judgment of the tester.

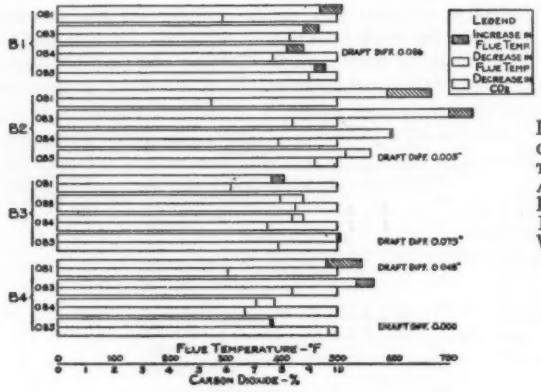

FIG. 25. COMPARISON OF FLUE TEMPERATURES OF SERIES *A* AND *C* AT THE SAME FUEL RATE. DRAFT DIFFERENCE 0.05 IN. WATER EXCEPT WHERE NOTED

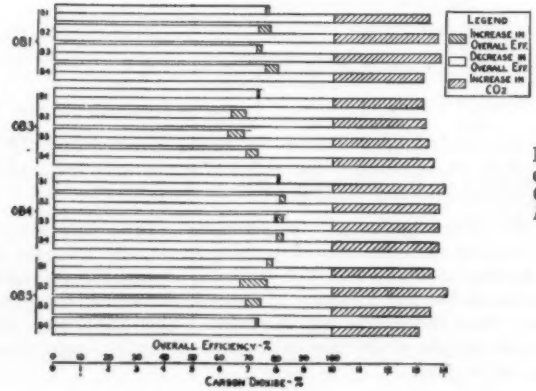

FIG. 26. COMPARISON OF EFFICIENCIES AND CO_2 OF SERIES *A* AND *D* AT THE SAME FUEL RATE

19. Fig. 26 shows increased CO_2 as high as 14.1 per cent and increased efficiencies in all but one case. It will be noted that the increase in efficiency for CO_2 above 10 per cent is not as great in proportion as the decrease in efficiency with decrease in CO_2 (See Fig. 24). It is evident from a practical standpoint that even though maximum CO_2 is ideal, there is a decreasing incentive to reach this ideal. A CO_2 value of 10 per cent was originally chosen for this work as a reasonable value to use. Furthermore, it is by no means

certain that for everyday use a very high CO_2 might not be productive of other difficulties which would more than offset the gain in efficiency.

20. Fig. 27 compared to Fig. 25 shows that with an increase in CO_2, flue gas temperatures generally show a decrease. This would be expected from a study of Fig. 25. It probably is true that combustion temperatures are higher in series *D* tests than in those of series *C*. Why should the flue temperatures show the reverse effect? The answer evidently is in the effect of combustion temperature on heat radiation rate in the furnace. The radiation rate does not increase directly with temperature, but as a power of the temperature commonly considered the fourth power. For example, a 200-deg. increase in combustion temperature may only be a 10 per cent increase, but this change in heat radiation rate may be 30 per cent. It is evident that the more heat is absorbed by radiation in the boiler furnace the less there will be for the heating surface in the convection zone to absorb.

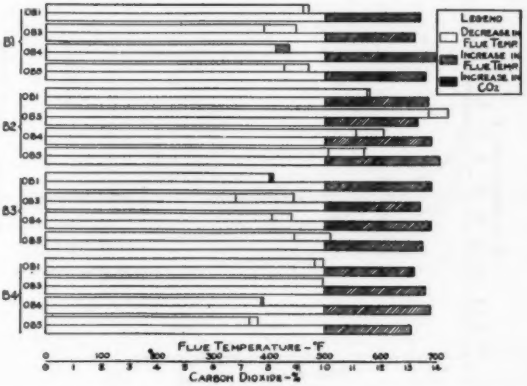

FIG. 27. COMPARISON OF FLUE TEMPERATURES OF SERIES *A* AND *D* AT THE SAME FUEL RATE

Series E: Series *E* represents the intermittent type of test mentioned in the proposed test code. A standard series *A* condition at about 1000 sq ft of steam "radiation" output is obtained and then various combinations of *on* and *off* periods are utilized to obtain varying average outputs. By this scheme an efficiency curve may be made.

21. Figs. 29 to 44 show the results, but it should be emphasized that the point of highest efficiency is about where the burner operates continuously. Due to losses during the *off* periods the overall efficiency will decrease more and more as the *off* period durations are longer and longer. The characteristic efficiency curve is the same shape for all boilers in contrast to the showing for series *A* (Fig. 5).

It is interesting to note that as a general fact the series *E* and *A* curves will not be greatly different if, when the burner is on, it is operating at or near the maximum efficiency of the boiler. Although a high operating efficiency is desirable, it is evident that average efficiencies will be less. The higher the *on* efficiency the higher will be the average, however. *OB 3—B 3* (Fig. 35) seems to contradict this, but it is probably an undetected error of some sort.

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22. It should be noted that *OB 5* may operate continuously by a graduated fire or by a *high* and *low* type of operation. Except for *OB 5—B 1* (Fig. 41), it may be said that either method will produce about the same result. *OB 5* as supplied gave 13 per cent CO_2 and it was necessary to modify certain air ports to decrease the CO_2. It is thought that a *high* and *low* with a 13 per cent CO_2 would improve the efficiency considerably.

It is of further interest to note that the CO_2 drops to low values at small outputs, which undoubtedly is one of the causes of the reduced efficiency, but in spite of this the performance from an efficiency standpoint is not greatly different from the *on* and *off* types of burners.

23. Fig. 28 shows cooling rate curves as typical of what happens as soon

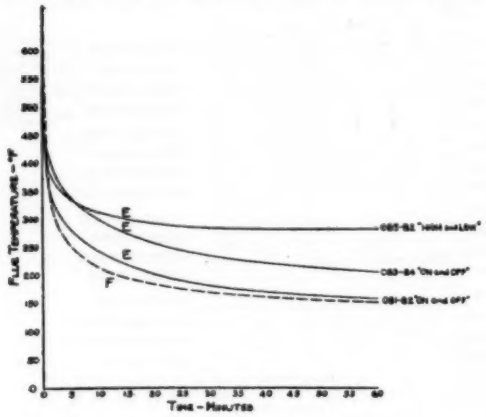

FIG. 28. CHANGE IN FLUE TEMPERATURE DURING AN *Off* PERIOD PLOTTED AGAINST TIME. (*E* REPRESENTS 0.02 IN. DRAFT SETTING AND *F* REPRESENTS 0.07 IN. DRAFT SETTING)

as the oil burner shuts off. Note that *OB 5* goes on *low* operation and that the curve flattens out at about 270 F.

Series F: Series *F* is similar to series *E* except that the furnace draft is increased to 0.07 in water and intermittent tests run again.

24. The results shown in Figs. 29 to 44 merely substantiate the results of series *C* tests as to loss of efficiency. Series *F* efficiency curves are similar to those of series *E* but more or less lower in value. It is evident that a dependable method of preventing the flow of air into and through the boiler during the *off* periods would do much to conserve heat. The presence of a fan does render a burner less sensitive to draft fluctuations and a high-pressure, high air velocity fan will afford greater resistance to air flow during *off* periods. Fig. 28 shows the difference in cooling rate time for *OB 1—B 2* for series *E* and *F*.

25. In all of the intermittent tests the *on* or *high* was for 30 min. No other combinations of *on* periods were tried. Thirty minutes were chosen because it was believed that this was fairly typical for residences thermostatically operated. If pressure regulation is used the number of *off* periods may be greater in number per 24 hr. If this is so the average operating efficiency is likely to be lower. It is hoped to investigate this point later.

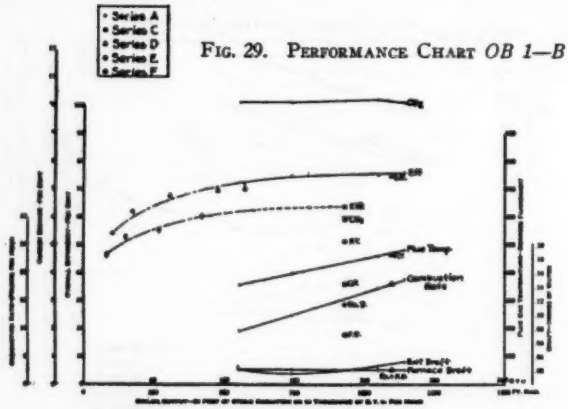

FIG. 29. PERFORMANCE CHART *OB 1—B 1*

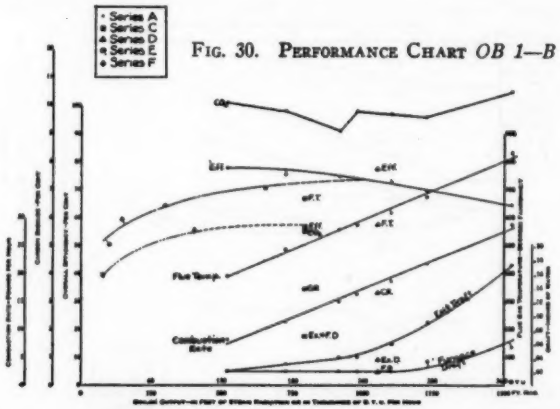

FIG. 30. PERFORMANCE CHART *OB 1—B 2*

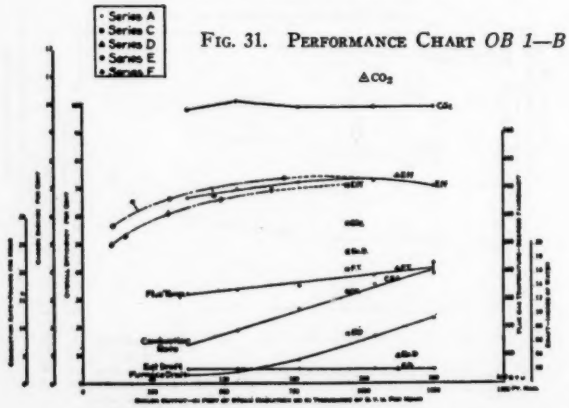

FIG. 31. PERFORMANCE CHART *OB 1—B 3*

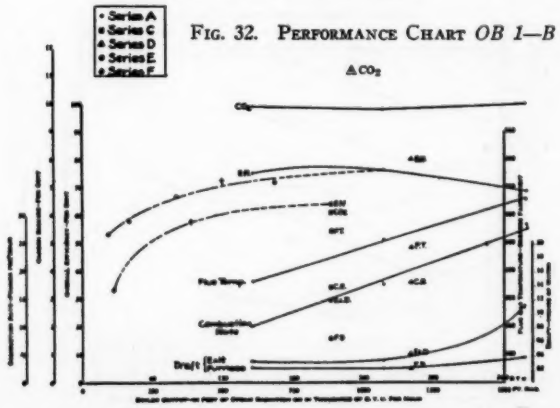

FIG. 32. PERFORMANCE CHART *OB 1—B 4*

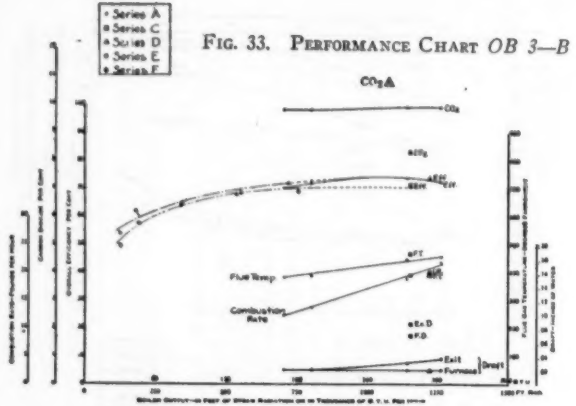

FIG. 33. PERFORMANCE CHART *OB 3—B 1*

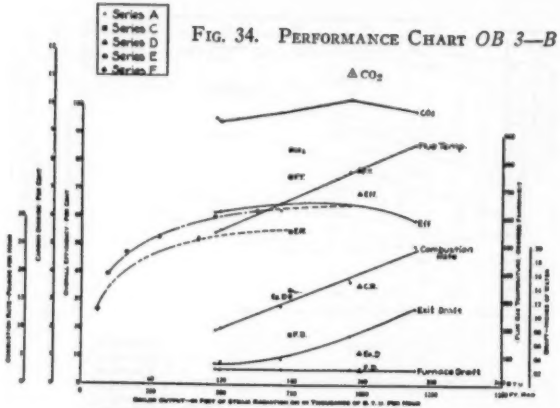

FIG. 34. PERFORMANCE CHART *OB 3—B 2*

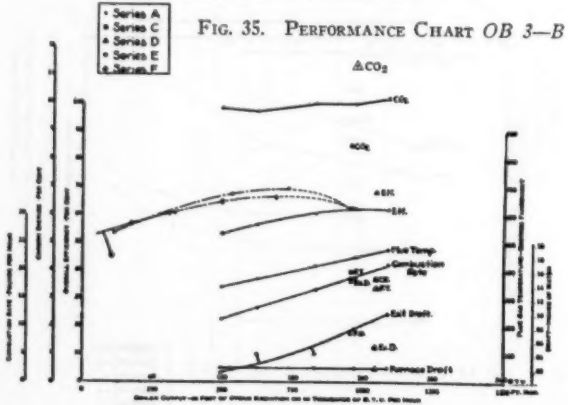

FIG. 35. PERFORMANCE CHART *OB 3—B 3*

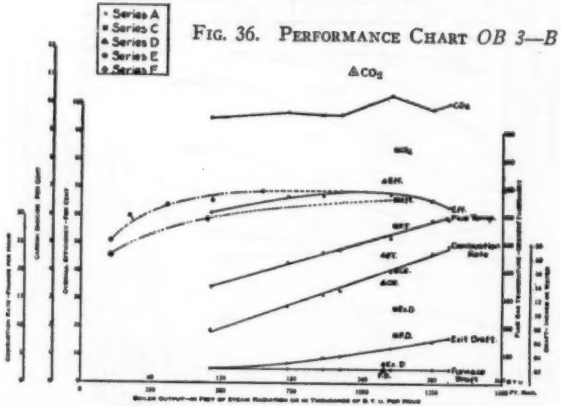

FIG. 36. PERFORMANCE CHART *OB 3—B 4*

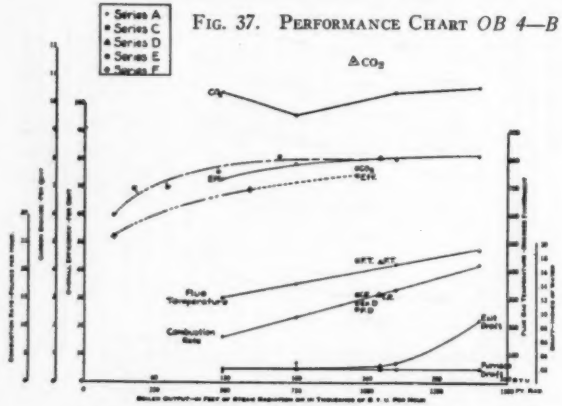

FIG. 37. PERFORMANCE CHART *OB 4—B 1*

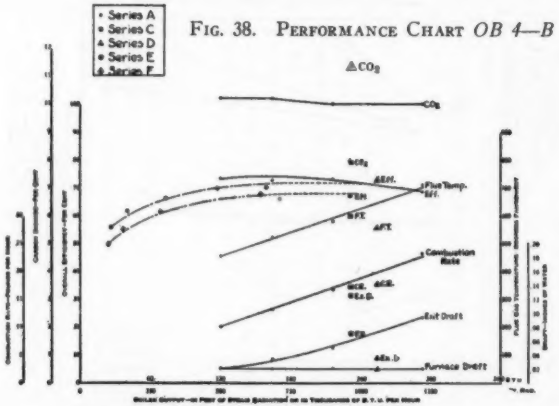

FIG. 38. PERFORMANCE CHART *OB 4—B 2*

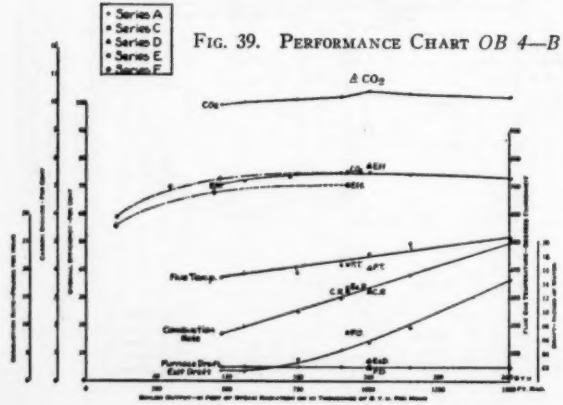

FIG. 39. PERFORMANCE CHART *OB 4—B 3*

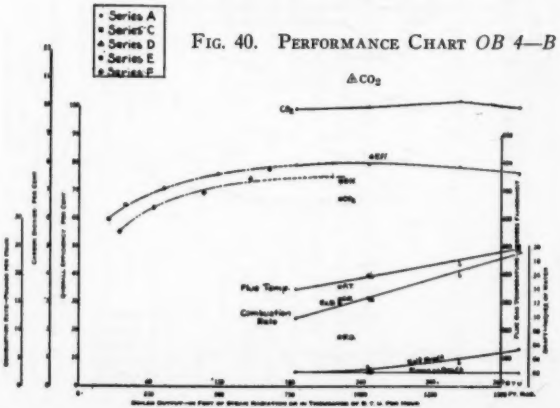

FIG. 40. PERFORMANCE CHART *OB 4—B 4*

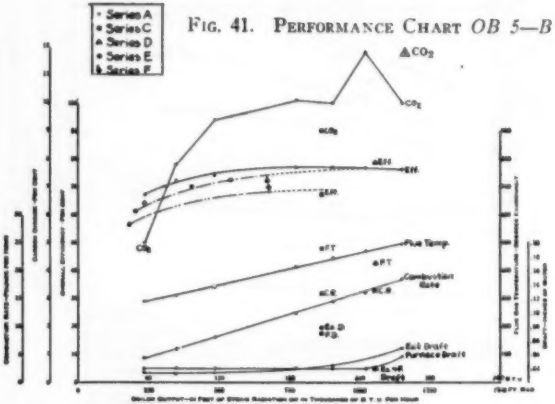

FIG. 41. PERFORMANCE CHART *OB 5—B 1*

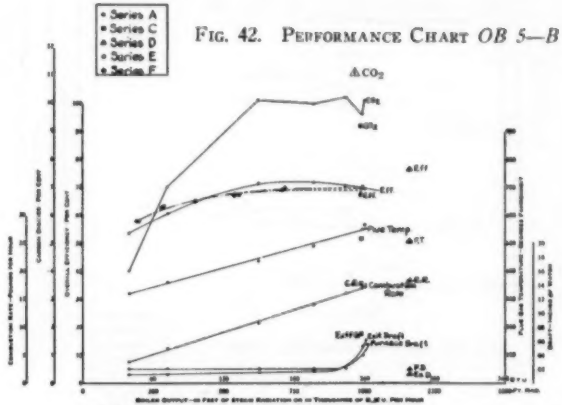

FIG. 42. PERFORMANCE CHART *OB 5—B 2*

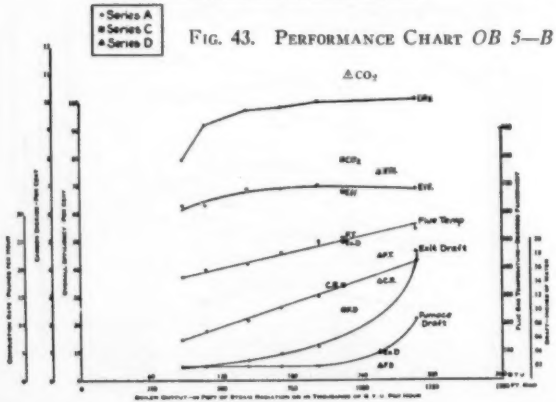

FIG. 43. PERFORMANCE CHART *OB 5—B 3*

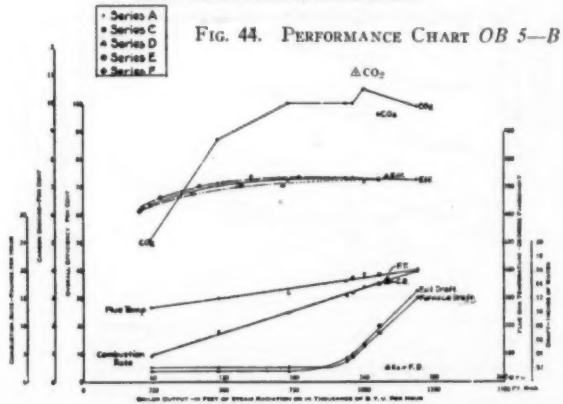

FIG. 44. PERFORMANCE CHART *OB 5—B 4*

26. The method of regulation and the type of burner (*i.e.*, continuous and intermittent) should receive future study as to sizing and adjusting for fuel rate in relation to the heating load. Pick-up load would be the important factor to consider. The characteristics of the boiler should likewise receive study in the same connection.

Series G: Series *G* were the sound tests reported in Section 13 in connection with series *A* results.

SUMMARY

(1) The general shape of the boiler efficiency curves remains the same with various oil burners. (2) The efficiency values for a given boiler are not necessarily equal for various oil burners even when the oil burners are operating in practically the same manner in each case. (3) The radiation and unaccounted losses average higher for some boilers than others. (4) Some burners also show higher radiation and heat losses than others. (5) All other things being equal an increase in draft: (*a*) lowers boiler efficiency (*b*) lowers CO_2 (*c*) increases excess air and (*d*) increases flue temperatures. (6) All other things being equal a decrease in excess air when CO_2 is 10 per cent; (*a*) increases CO_2 (*b*) increases efficiencies moderately and (*c*) decreases flue temperatures. (7) Efficiency curves on intermittent type of operation all show the same characteristic shape. (8) Intermittent operation at different drafts shows that some burners are more sensitive to draft fluctuations than others. Item (5) also indicates the same characteristic. (9) Efficiencies tend to rise with a decrease in heat absorption rate of boiler per square foot of heating surface although some types of burners will give good results at higher rates than others. (10) Increase in boiler heating surface for the same fuel burning rates will show higher efficiencies. (11) Heat liberation rates in furnaces vary quite widely, averaging higher for some oil burners than others. There seems to be no such thing as a desirable fuel burning rate per cubic foot of furnace volume for oil burners in general. (12) On sound tests some burners make measurable combustion sounds but it does not necessarily follow that such a burner may not be more quiet than one giving no combustion sounds at all.

DISCUSSION

W. F. KLOCKAU (WRITTEN): This paper deserves the highest praise.

Nothing is said about the construction of the combustion chambers, especially for the gun type burners. As the form of the combustion chamber can materially affect the performance and efficiency of a gun type burner, the writer believes that before any definite statement can be made regarding the merits of this type of burner that the form of the combustion chamber recommended by the manufacturer should be used in making the tests.

The design of some gun type oil burners is such that it is necessary for the combustion chamber to assist in mixing the air and oil. It readily can be understood that if a gun type burner is placed in a rectangular combustion chamber and then, without any other adjustments, placed in a combustion chamber of a form recommended by the manufacturer, the efficiency immediately is changed as well as the character and color of the flame.

J. T. WEBBER (WRITTEN): The statement, "the results reported in this paper indicate the performance that would logically be expected under the usual operating

conditions" is true for laboratory tests but I believe that mention should be made here of the fact that radiation losses in laboratory tests usually are utilized in residence heating.

In relation to the use of No. 1 Domestic Fuel Oil, Light in these tests it should be acknowledged that this grade was the maximum for some but not all of the burners under listings of the Underwriters Laboratories. Possibly it would be well to give maximum grades of oil with which the various burners can be used under their listings.

No. 908

HEAT AND MOISTURE LOSSES FROM MEN AT WORK AND APPLICATION TO AIR CONDITIONING PROBLEMS

By F. C. HOUGHTEN [1] *(MEMBER)*, W. W. TEAGUE [2] *(MEMBER)*, W. E. MILLER [3] *(NON-MEMBER)*, AND W. P. YANT [4] *(NON-MEMBER)*. PITTSBURGH, PA.

HEAT and moisture dissipation from the human body, for various degrees of physical exertion and for a wide range of atmospheric conditions, has been the subject of investigation at the Research Laboratory of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS for the past four years. Reports giving the relation of the temperature, humidity, and air motion of the atmosphere to the rate of heat and moisture dissipation from the bodies of normally clothed men seated at rest were published two years ago[a,b]. The Laboratory has since published a paper[c] emphasizing the physiological and hygienic aspects of heat and moisture dissipation from the bodies of men working in still air, which paper deals with the practical application of these data to air conditioning problems. The findings of the Laboratory for men at rest found immediate application to the air conditioning and cooling of theaters, auditoriums, and other places of assemblage where large numbers of persons are seated at rest.

The results outlined herewith may find similar application to the conditioning of air in dance halls and other places of assemblage where people are active, and in industrial plants where the highest efficiency and the comfort of workers are desirable.

PRACTICAL DATA AND THEIR APPLICATION TO AIR CONDITIONING PROBLEMS

In the solution of problems involving air conditioning of space occupied by persons working or otherwise engaged in physical exertion the engineer is

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[4] Supervising Chemist, Health Laboratories Section, U. S. Bureau of Mines.

[a b c] See Bibliography.

Presented at the Semi-annual Meeting of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, Swampscott, Mass., June, 1931, by F. C. Houghten.

interested in the total rate of heat dissipation to the atmosphere by the average person. He is also interested in the rate of sensible and latent heat dissipation for all atmospheric conditions met within practice and the rate at which moisture is added to the atmosphere. In the solution of certain problems he is interested in knowing the percentage of the total loss which is dissipated as sensible and latent heat. These data for men working at three rates are given in convenient form for practical application in Figs. 1, 2, 3 and 4. Similar data for men at rest are also given for comparison.

Although total heat loss and sensible and latent heat losses are not exact functions of effective temperature and dry-bulb temperature respectively for all conditions of humidity, they are plotted as such in the curves Figs. 1 to 4. This is accomplished by approximations not always rigidly accurate but sufficiently so for application to most practical problems. As will be shown later by the curves in Figs. 14 and 15, the latent and sensible heat losses show some variation with change in humidity for dry-bulb temperatures above 75 F.

The curves for men working at 33,075 ft-lb per hour were established by a large number of points at various temperatures and two humidities, this being the rate of work engaged in for the majority of the tests. The curves for work rates of 16,538 and 66,150 ft-lb per hour respectively were drawn from data on four men at an effective temperature of 70 F only and the application of the established relationship between the curves for men seated at rest and working at 33,075 ft-lb per hour. Notwithstanding these approximations and interpolations, the data in Figs. 1 to 4 are sufficiently accurate for most practical application. Where extreme accuracy is desired, particularly for definite humidities at high temperatures, it is recommended that the variation in the data for different atmospheric conditions as given in Figs. 11 to 17 be taken into consideration.

Figs. 1 to 4 may be used in the solution of problems involving other types and rates of physical exertion than those for which the curves are drawn, provided the total rate of energy transformation within the body or the metabolic rate, and the rate at which energy is expended in performing external work which is not transformed into heat is known. Table 1 gives the metabolic rates for various degrees of activity. The metabolic rate for other types of activity may be estimated with the aid of the values in this table or they may be determined by independent metabolism tests.

By plotting the metabolic rate (or this rate less the rate of expending energy in external work) as point x, against the proper effective temperature in Fig. 1, a curve E may be drawn through this point similar to curves A, B, C and D. Curve E will then give the relation between total heat loss and effective temperature for this new rate of work. Likewise, curve E may be drawn in Figs. 2 and 3 bearing the same relation to curves A, B, C and D in these figures as the curve E does to these respective curves in Fig. 1.

In walking and many other types of physical exertion practically all the energy shown by metabolic tests, is transformed into heat within the body and external work need not be considered. As will be shown later in Fig. 10, the human body at best is extremely inefficient as an engine for performing external mechanical work and as a result the greater part of the energy transformation within the body as determined by metabolism becomes heat which must be dissipated to the air in either the sensible or latent form. For the type of work

engaged in for this study, 11 per cent of the energy shown by metabolism left the body for external work. It is probable that few types of work will show a greater mechanical efficiency. Hence for most purposes rate of total heat dissipation may be taken as equal to the metabolic rate. If a closer ap-

TABLE 1. RELATION BETWEEN METABOLIC RATE AND ACTIVITY ACCORDING TO VARIOUS AUTHORITIES

Activity	Metabolic Rate		
	Cal per Sq M per Hour	Btu per Hour for Avg. Man (19.5 Sq Ft Surface Area)	Authority
Seated at rest	53.4	384	*A*
Standing at rest	60	431	*A*
Going through motions necessary to operate work machine 7.5 times per minute	98	704	*A*
Operating work machine 7.5 times per minute, raising weight only	145	1042	*A*
Operating work machine 7.5 times per minute	118	848	*A*
Operating work machine 3.75 times per minute	96	690	*A*
Operating work machine 15 times per minute	180	1293	*A*
Walking 2 mph	106	761	*B*
Walking 3 mph	146	1049	*C*
Walking 4 mph	193	1388	*B*
Walking 4½ mph	241	1731	*C*
Walking 5 mph	352	2530	*C*
Slow run	318	2285	*D*
Maximum exertion (without food)	464–663+	3333–4762+	*D*
Level walking 2.2 mph	111	797	*E*
10 per cent grade 2.4 mph	233	1675	*E*
25 per cent grade 2.0 mph	414	2975	*E*
40 per cent grade 2.4 mph	559	4018	*E*
Very severe exercise	356	2555	*F*
Exercising more or less severely	85	610	*G*
Tailor	67	482	*H*
Bookbinder	87	626	*H*
Shoemaker	92	661	*H*
Carpenter	106–134	762–963	*H*
Metal worker	120	862	*H*
Painter (of furniture)	122	876	*H*
Stonemason	207	1488	*H*
Man sawing wood	250	1797	*H*

Authorities

A Research Laboratory, AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS.
B Average values from (Douglas, Haldane, Henderson, and Schneider) and (Henderson and Haggard).
C Douglas, Haldane, Henderson and Schneider.
D Henderson and Haggard.
E H. M. Smith.
F Benedict and Carpenter.
G Atwater and Benedict.
H Becker and Hamalainen.

proximation is desired, the efficiency of the body in performing such work must be estimated or determined by test.

EXAMPLES IN THE USE OF HEAT AND MOISTURE LOSS DATA

Typical applications of the data given in Figs. 1 to 4 and in Table 1 are shown in the solution of the following problems:

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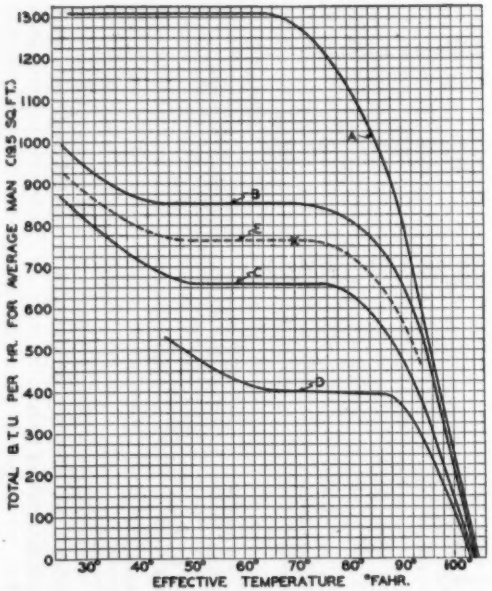


FIG. 1. RELATION BETWEEN TOTAL ENERGY LOSS FROM THE HUMAN BODY AND EFFECTIVE TEMPERATURE FOR STILL AIR *

* Curve *A*—Men working, 66,160 ft-lb per hour. Curve *B*—Men working 33,075 ft-lb per hour. Curve *C*—Men working, 16,538 ft-lb per hour. Curve *D*—Men seated at rest. Curves *A* and *C* drawn from data at an effective temperature of 70 deg. only and extrapolating the relation between curves *B* and *D*, which were drawn from data at many temperatures.

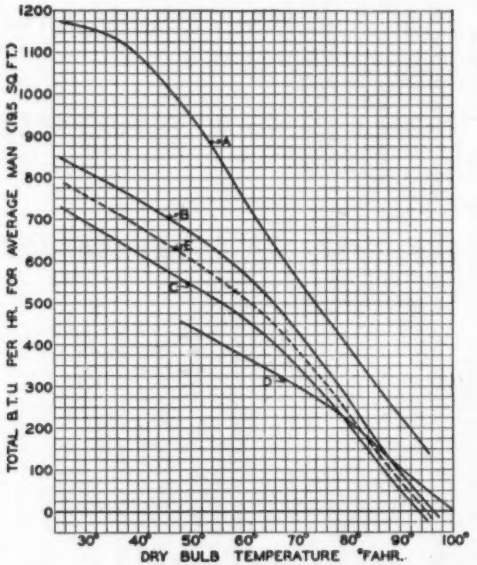


FIG. 2. RELATION BETWEEN SENSIBLE ENERGY LOSS FROM THE HUMAN BODY AND DRY-BULB TEMPERATURE FOR STILL AIR *

* Curve *A*—Men working, 66,150 ft-lb per hour. Curve *B*—Men working, 33,075 ft-lb per hour. Curve *C*—Men working, 16,538 ft-lb per hour. Curve *D*—Men seated at rest. Curves *A* and *C* drawn from data at a dry-bulb temperature of 81.3 F only and extrapolating the relation between curves *B* and *D* which were drawn from data at many temperatures.

Problem 1a. How much sensible heat, how much latent heat and how much water vapor will be added to the atmosphere per hour by 1,000 men while working at the rate of 33,075 ft-lb per hour in an atmosphere of 50 F dry-bulb and 45.5 F wet-bulb, respectively.

b. If the dry and wet-bulb temperatures were 75 and 59 F, respectively, how much heat and moisture would be dissipated to the atmosphere?

c. How do the losses in (*b*) compare with the same losses for 1,000 men seated at rest?

Solution—Problem 1a: From Fig. 2, curve *B*, find the sensible heat loss per person for a dry-bulb temperature of 50 F to be 670 Btu per hour. From Fig. 3 find the latent heat loss per person for a dry-bulb temperature of 50 F to be 165 Btu per hour and the moisture added to be 1,150 grains per hour. $1{,}000 \times 670 = 670{,}000$ Btu sensible heat; $1{,}000 \times 165 = 165{,}000$ Btu latent heat, and $1{,}000 \times 1{,}150 = 1{,}150{,}000$ grains, or 164 lb of water per hour will be added.

Solution—Problem 1b: From Figs. 2 and 3, respectively, curve *B*, the sensible and latent heat losses per person for a dry-bulb temperature of 75 F are found to be 360 and 515 Btu per hour. The water vapor added to the atmosphere is 3,480 grains per hour. The working men will then add 360,000 Btu sensible heat, 515,000 Btu latent heat and 3,480,000 grains or 497 lb of water vapor to the air per hour.

Solution—Problem 1c: From Figs. 2 and 3, respectively, curve *D*, find the sensible and latent heat losses at a dry-bulb temperature of 75 F for a person seated at rest to be 265 and 135 Btu per hour. The water vapor added to the air will be 900 grains per hour. The men will then add to the atmosphere 265,000 Btu sensible heat, 135,000 Btu latent heat and 900,000 grains or 129 lb of water vapor.

From Solution (*b*), $360{,}000 - 265{,}000 = 95{,}000$ Btu more sensible heat for men working at 33,075 ft-lb per hour than for men seated at rest. Hence, $515{,}000 - 135{,}000 = 380{,}000$ Btu per hour more latent heat. Therefore, $497 - 129 = 368$ lb per hour more water vapor.

Problem 2. Neglecting the gain or loss of heat to a building by transmission or infiltration through the walls, windows and doors, how many cubic feet of outside air with dry-bulb, wet-bulb and effective temperatures of 65 F, 59 F and 62.8 F, respectively, must be added per hour to a building containing 1,000 persons working at the rate of 33,075 ft-lb per hour in order that the inside dry and wet-bulb temperatures shall not exceed 75 and 65 F, respectively?

Solution—Problem 2: Figs. 2 and 3 gives 360 Btu sensible heat and 3,480 grains of moisture as the additions per person with dry-bulb temperatures of 70 F in the building. Therefore, 360,000 Btu of sensible heat and 3,480,000 grains of moisture will be added to the air per hour.

Taking 0.24 as the specific heat of air, 2.4 Btu per pound of air will be required to raise the dry-bulb temperature from 65 to 75 F and $\frac{360{,}000}{2.4} = 150{,}000$ lb of air or $150{,}000 \times 13.4 = 2{,}010{,}000$ cu ft of air per hour will be required. This is equivalent to $\frac{2{,}010{,}000}{60 \times 1{,}000}$ or 33.5 cu ft per person per minute.

The moisture content of the inside air as taken from a psychrometric chart is 76 grains per pound of dry air and that of the outside condition 65 grains, so the increase in moisture content will be 11 grains per pound of dry air; $\frac{3{,}480{,}000}{11}$ or 316,400 lb of dry air, or approximately 320,000 lb of air at the specified condition, will be required. This is equivalent to $320{,}000 \times 13.4 = 4{,}290{,}000$ cu ft of air per hour or $\frac{4{,}290{,}000}{60 \times 1{,}000} = 71.5$ cu ft of air per minute per person.

The higher volume of 71.5 cu ft per person per minute will be required to keep the wet-bulb temperature from rising above the 65 F specified. The dry-bulb temperature will, therefore, not rise to the maximum of 75 F.

XUM

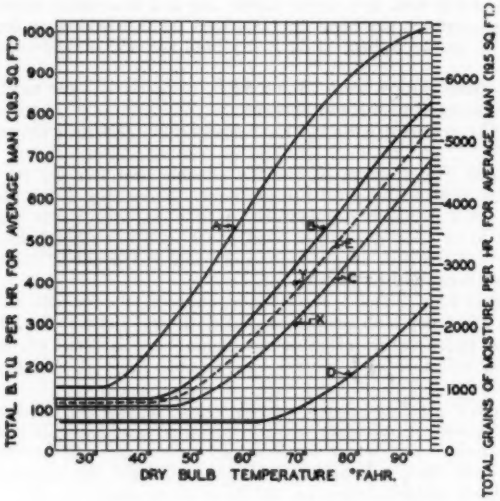

TOTAL B.T.U. PER HR. FOR AVERAGE MAN (185 SQ. FT.)
TOTAL GRAINS OF MOISTURE PER HR. FOR AVERAGE MAN (185 SQ. FT.)
DRY BULB TEMPERATURE °FAHR.
A
B
C
D

FIG. 3. RELATION BETWEEN HEAT AND WEIGHT LOSS FROM THE HUMAN BODY BY EVAPORATION AND DRY-BULB TEMPERATURE FOR STILL AIR *

* Curve *A* — Men working, 66,150 ft-lb per hour. Curve *B* —Men working, 33,075 ft-lb per hour. Curve *C*—Men working, 16,538 ft-lb per hour. Curve *D*—Men seated at rest. Curves *A* and *C* drawn from data at a dry-bulb temperature of 81.3 F only and extrapolating the relation between curves *B* and *D* which were drawn from data at many temperatures.

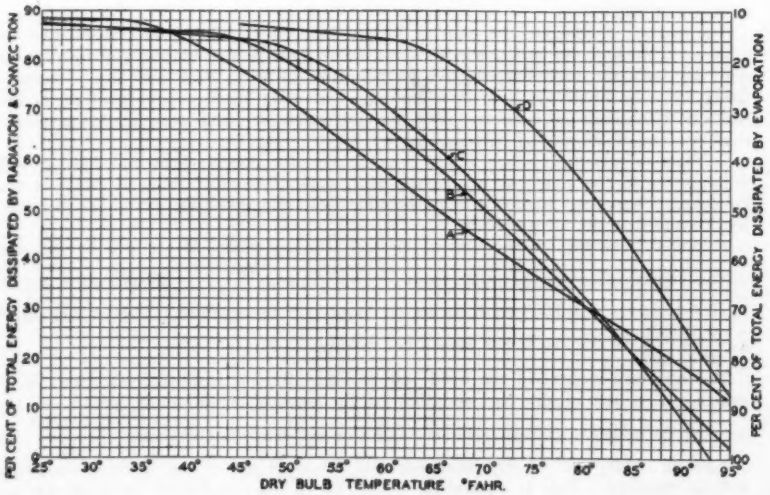


FIG. 4. RELATION BETWEEN ENERGY LOSS FROM THE HUMAN BODY BY EVAPORATION, RADIATION AND CONVECTION, AND DRY-BULB TEMPERATURE FOR STILL AIR *

* Curve *A*—Men working, 66,150 ft-lb per hour. Curve *B*—Men working, 33,075 ft-lb per hour. Curve *C*—Men working, 16,538 ft-lb per hour. Curve *D*—Men seated at rest. Curves *A* and *C* drawn from data at a dry-bulb temperature of 81.3 only and extrapolating the relation between curves *B* and *D* which were drawn from data at many temperatures.

Problem 3. A subway station has an average capacity of 5,000 persons. Assuming that the average state of activity of the crowd is walking 2 miles per hour, how much heat and moisture will be added to the atmosphere if the dry- and wet-bulb temperatures are maintained at 80 F and 66 F, respectively?

Solution—Problem 3: From Table 1 find the metabolic rate for walking 2 miles per hour to be 761 Btu per hour. Plotting this value in Fig. 1 at the constant portion of the curve and extrapolating the relation between curves *A, B, C* and *D*, dotted curve *E* is drawn for walking 2 miles per hour. Applying this same relationship in Figs. 2 and 3 gives curves for sensible and latent heat losses at the given state of activity.

From Figs. 2 and 3 find the sensible and latent heat losses per person at a dry-bulb temperature of 80 F to be 240 and 530 Btu per hour. The water vapor added will be 3,580 grains per hour. The occupants of the station will then add 5,000 ×

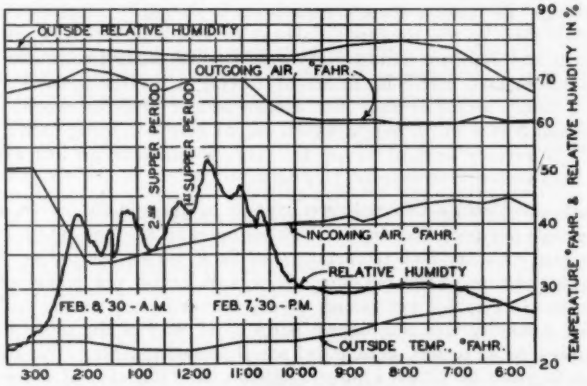


FIG. 5. TEMPERATURE AND HUMIDITY CHART FOR GREAT HALL THE MEMORIAL UNION BUILDING, UNIVERSITY OF WISCONSIN, FOR NIGHT OF FEB. 7-8, 1930

240 or 1,200,000 Btu sensible heat, 5,000 × 530 or 2,650,000 Btu latent heat and 5,000 × 3,580 = 17,900,000 grains or 2,557 lb of water vapor to the air per hour.

HEAT AND MOISTURE LOSSES WHILE DANCING

A graph of temperature of supply and exhaust air, and inside and outside relative humidity taken during the Junior Prom in Great Hall Memorial Union Building, University of Wisconsin, is shown in Fig. 5. One thousand persons are estimated to have danced at the Junior Prom. This graph was supplied by Prof. G. L. Larson and is based on data collected by him in a study of ventilation requirements of the building and published by Prof. C. Braatz in The Wisconsin Engineer[d].

No heat or moisture was added to the atmosphere in the hall by the ventilating system during the dance. Hence the temperature and humidity rises are due to heat and moisture losses of those in attendance. The heating and humidifying effect of the dancers is clearly shown by the graph as pointed out by the authors of the university report as follows:

[d] See bibliography.

The inside relative humidity curve indicates that the Grand March started about 9:45, at which time the relative humidity began to climb slowly for about twenty minutes, then dropped off momentarily during the picture taking ceremony, which was followed by a very decided rise during the first dance. From here on each of the following dances has a decided hump of its own on the relative humidity curve. Note also that while the temperature of the incoming air gradually decreased throughout the entire evening, the temperature of the air leaving the hall rose from about 61 F to 70 F during the first dance and remained practically constant for the remainder of the evening.

It is of interest to compare the calculated rates of latent heat and moisture addition to the hall as given by the authors of the university report, or calculated from their graph, with the values established in this report for persons working. In making these comparisons it should be emphasized that the values established by the Research Laboratory and here reported are for men and the heat and moisture units are given in English units for an average sized man or a 150-lb man 5 ft 8 in. in height, having a surface area of 19.5 sq ft. Perhaps a more correct average for women college students would be 120 lb in weight and 5 ft 4 in. in height which would give according to the chart by Du Bois[a-b] a surface area of 17 sq ft. This would give an average surface area for both men and women of 18.2 sq ft.

The authors calculate the moisture content of the outside or incoming air to be 12 grains per pound of dry air. Based upon an average relative humidity of 42 per cent at a dry-bulb temperature of 70 F for the period of the dance, the average moisture of the outgoing air was 46 grains per pound of dry air, or 34 grains were added per pound of dry air delivered to the hall. When the relative humidity was the highest point at 11:40 PM the moisture content was 57 grains, or 45 grains were added per pound of dry air. The air delivery to the hall is given as 12,300 cfm or 738,000 cfh. The rate of moisture addition for the average relative humidity of 42 per cent and the highest relative humidity of 52 per cent is given by the equation

$$\text{Average rate} = \frac{738{,}000 \times 0.075 \times 34}{7{,}000} = 268.8 \text{ lb per hour.}$$

$$\text{Maximum rate} = \frac{738{,}000 \times 0.075 \times 45}{7{,}000} = 355.8 \text{ lb per hour.}$$

where 0.075 = weight per cubic foot of 70 F air.
and 7,000 = grains per pound.

The moisture addition per person is $\frac{268.8}{1000} = 0.2688$ and $\frac{355.8}{1000} = 0.3558$ lb per hour, and the latent heat added per person is $0.2688 \times 1036 = 278.5$ and $0.3558 \times 1036 = 368.6$ Btu per hour for the average and maximum rates, respectively. The rate of latent heat addition per hour per unit of surface area used in the charts in this report, or per 19.5 sq ft of body surface, is $278.6 \times \frac{19.5}{18.2} = 298.7$ Btu for the average time of the dance, and $368.5 \times \frac{19.5}{18.2} = 395.0$ Btu for the maximum rate. These two points are indicated by points x and y, respectively, Fig. 3. The maximum rate point y shows a rate of latent heat elimination slightly less than that for men working at the rate of 33,075 ft-lb per hour, or the average dancer probably expended energy at nearly at high a rate as he would while working at the rate of 33,075 ft-lb per

hour on the Laboratory machine. By a similar comparison the average rate of energy dissipation by the dancers for the whole evening, including intermission and supper period, was about equal to that of a man working at the rate of 16,538 ft-lb per hour.

The abrupt rise and fall in the relative humidity curve with the dance activity compared with the relatively uniform dry-bulb temperature rise of the air bears out the findings of the Laboratory that sensible heat loss is little affected by activity or other factors for the same dry-bulb temperature and that the body controls latent heat dissipation so as to make the sum of sensible and latent heat loss equal to heat production in order to maintain the proper body temperature.

TEST PROCEDURE AND OBSERVATIONS

The results reported in this paper were obtained by making 135 heat balance tests on five subjects in various controlled atmospheric conditions in the psychrometric chambers of the Research Laboratory of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS in the Pittsburgh Station of the U. S. Bureau of Mines. These two rooms, in either one of which any desired atmospheric condition could be had and accurately controlled, are described in another Laboratory report[f]. The heat balance test procedure was in general similar to that published in an earlier Laboratory report[a,b]. Briefly, the subjects were made to work at a uniform rate in the desired atmospheric condition. Their weight loss and the consumption of O_2 were determined for the period of work, and from these observations the rate of oxidation of food, the rate of production and exhalation of CO_2, the rate of weight loss due to exhalation of carbon in the CO_2 produced and exhaled, the rate of weight loss due to evaporation of perspiration from the body surface and the respiratory tract and the rate of heat production in the body by metabolism were calculated. From these values and the change in body temperature, the total rate of heat dissipation to the atmosphere and its differentiation into sensible and latent heat losses were calculated.

Since energy production or metabolic rate varies to some extent with the diet of the subject the tests were made at various times during the day in reference to meals in order that the results might apply as nearly as possible to average working conditions.

In order that the results might be directly comparable with those reported[a,b] earlier for persons seated at rest without the introduction of other variables than the degree of activity, the clothing worn by the subjects was the same as that worn by the subjects in the earlier investigation. They were clothed in light athletic underwear, light-weight socks, low cut shoes, shirt with collar attached, and wool-cotton mixed business suits kept at the Laboratory for the purpose of such tests.

In reporting for the tests the subjects changed clothing, and then entered the air-conditioned room and started working. After a 30-min preliminary work period the test was started and continued for one hour, excepting in tests at an effective temperature of 90 F when the test was shortened to 15-min preliminary and 30-min actual test periods because of the inability of the men to endure the high temperatures for a longer period.

[a b f] See bibliography.

Body temperatures (rectal) were taken at the beginning of the preliminary test period and at the beginning and end of the test period. These temperatures were needed in order to calculate the amount of heat storage in the body because of the heat capacity and change of temperature of the body during the test.

The subjects were weighed on the bullion balance (Fig. 6) to an accuracy of 0.2 gram at the beginning and end of the test period[a]. The weight loss was due to evaporation of moisture from the body surface and clothing and from

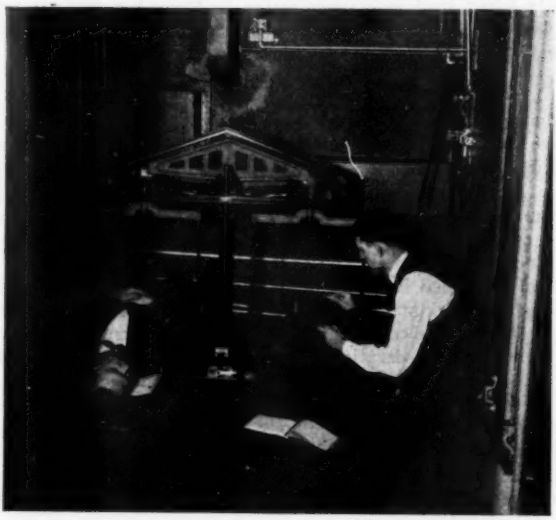


FIG. 6. BALANCE ON WHICH SUBJECTS ARE WEIGHED

the respiratory tract, and also due to the CO_2 exhaled less the O_2 consumed. The rate of O_2 consumption and CO_2 production were calculated from the O_2 consumption, as determined on a new model Benedict-Roth Recording Metabolism apparatus (Fig. 7), the operation of which is explained by Du Bois[e]. A spirometer is filled with pure O_2, from which the subject inhales through a mouth-piece, exhaling back into a closed system through a cannister of soda lime, which removes the CO_2 formed by the oxidation of food in the body. The drop of the spirometer bell indicates the volume of O_2 consumed and is recorded on a kymograph drum. The instrument is calibrated to read directly in calories, a unit of energy production within the body resulting from the oxidation of food, assuming an average respiratory quotient of 0.82, according to Benedict, Emmes, Roth and Smith[b]. This recorded value when corrected for the prevailing temperature of the O_2 in the spirometer and barometric pressure give the rate of energy production or metabolic rate. There is evidence reported by Smith[i] that under some extreme conditions of work which required the same O_2 consumption as that reported in this paper, the respiratory

[a e b i] See bibliography.

quotient may be as high as 0.92. If this higher value had been used it would have increased the rate of heat production by 3 cal per square meter per hour, the heat loss by radiation and convection by 6 cal, and decreased the calculated heat loss by evaporation by 3 cal.

Total rate of heat loss, loss by radiation and convection combined and loss by evaporation were calculated as described in an earlier laboratory report[a]

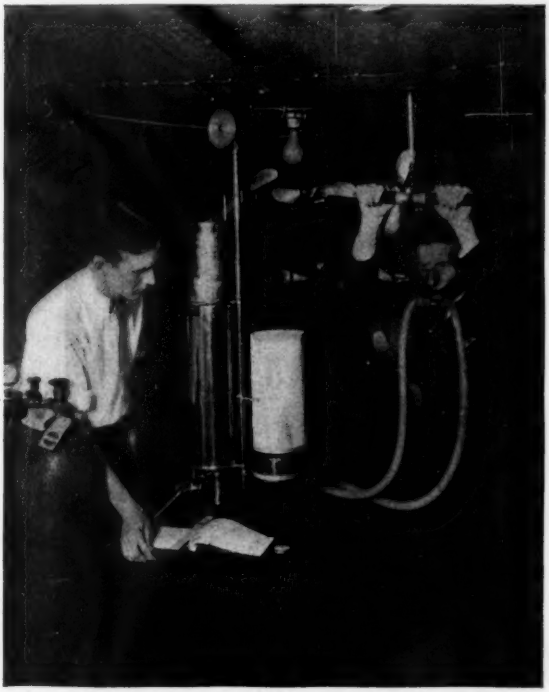


FIG. 7. BENEDICT-ROTH METABOLISM APPARATUS FOR DETERMINING METABOLIC RATE

from the metabolic rate, weight loss and body temperature rise. Briefly, the total energy loss was taken as the rate of energy production minus the rate of heat storage in the body due to its temperature rise and specific heat (0.83). The total heat loss was taken as the total energy loss minus the mechanical energy expended on the work machine. Heat loss by evaporation or the latent heat loss was computed from the weight loss by evaporation, and the heat of evaporization of water at body temperature. The radiation and convection, or sensible heat loss, was taken as the total heat loss minus the loss by evaporation. For convenience in the use of the instruments available the observations and

[a] See bibliography.

most of the calculations were made in the metric system of units and the results were later translated into the English units for the convenience of the engineer.

The work was performed on a special work machine (Fig. 8), consisting of a 37-lb weight which was raised and lowered through a distance of 4 ft 10.8 in. by a rope over a pulley. In the majority of the tests, the weight was raised and lowered once every 8 seconds. The energy which was expended on the machine in raising the weight was taken as the product of the force necessary to raise the weight at the customary rate (44.5 lb), the distance through which

FIG. 8. WORK MACHINE

it was raised (4 ft 10.8 in.), and the heat equivalent of one foot-pound of work (0.001285 Btu). The energy expended on the machine in raising and lowering the weight was taken as the product of the force necessary to raise the weight minus the force necessary to retard the fall to the desired rate, the distance through which it was raised, and the heat equivalent of work.

The psychrometric chart (Fig. 9), with effective temperature lines for individuals normally clothed working at the rate of 33,075 ft-lb per hour in still air, was used in analyzing the data presented. This chart was developed at the Research Laboratory and is very similar to other charts published by the Laboratory. The effective temperature lines in this chart have a slightly greater slope than those for individuals at rest[j], but a slightly lesser slope than those for individuals lightly clothed, working[k].

The data collected are for effective temperatures ranging from 27 to 90 deg at relative humidities of approximately 20 and 95 per cent. The limits of the

j k See bibliography

effective temperature range were chosen as the extremes at which individuals could work at the prescribed rate in the clothing worn, without excessive discomfort.

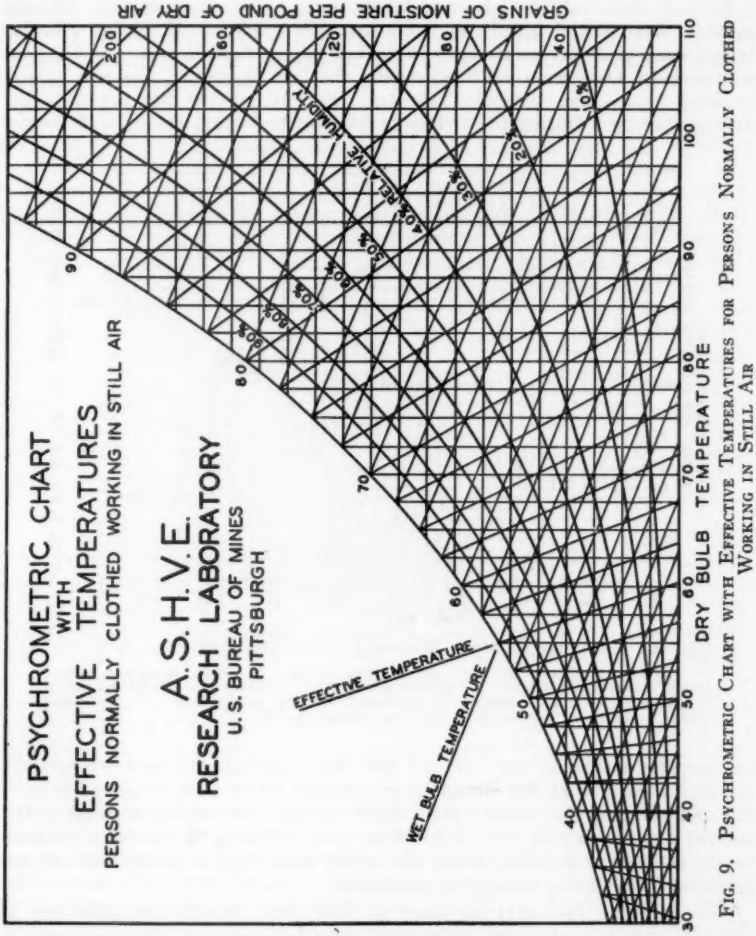


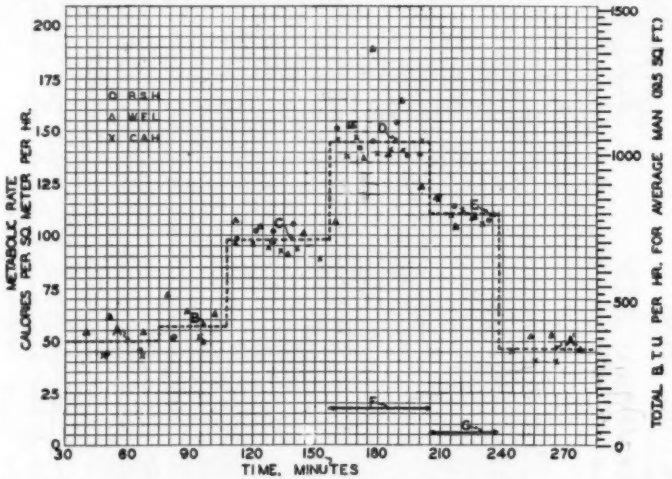
FIG. 9. PSYCHROMETRIC CHART WITH EFFECTIVE TEMPERATURES FOR PERSONS NORMALLY CLOTHED WORKING IN STILL AIR

DATA AND RESULTS

The relation between metabolism and degree of physical activity is shown in Fig. 10. In this chart and those which follow, all temperatures are expressed in both degrees centigrade and degrees Fahrenheit. Heat loss values are expressed in calories per square meter of body surface per hour as is the accepted practice of the physiologist, and also in English units as Btu per hour for

"a unit of surface area equal to that of an average sized man," that is, a 150-lb man, 5 ft 8 in. in height having a surface area of 19.5 sq ft or 1.81 sq m. Surface areas were determined from the chart by Du Bois*[b].

Each of three subjects whose metabolic rates are distinguished by different symbols were seated quietly in a constant air condition of 70-deg effective temperature at 20 per cent relative humidity for the first 75 min of a 4¾ hour test period. Data were collected during the last 45 min of the rest period, showing an average metabolic rate (*A*), Fig. 10, of 359 Btu per hour per average person. During the following 33 min they stood quietly and showed



—FIG. 10. RELATION BETWEEN METABOLIC RATE AND DEGREE OF ACTIVITY *

* *A* and *A'*—Seated at rest. *B*—Standing quietly. *C*—Body movement without raising and lowering weight. *D*—Raising weight only. *E*—Raising and lowering weight. *F*—Mechanical energy expended on machine in raising weight only. *G*—Mechanical energy expended on machine in raising and lowering weight.

an average metabolic rate (*B*) of 409 Btu. During the next 50 min the subjects went through the customary movements of the body employed in raising and lowering the weight once every 8 seconds but without moving it and showed a metabolic rate (*C*) of 704 Btu. The following 48 min were occupied by the subjects in actually raising the weight once every 8 seconds but without lowering it, this being done by an attendant.

The metabolic rate (*D*) increased to 1042 Btu. During the following 33 min the subjects again went through the same body motions, this time actually raising and lowering the weight at the previous rate. Their metabolic rate (*E*) decreased to 790 Btu. During the last 45 min the subjects were again seated quietly and showed a metabolic rate (*A'*) of 338 Btu per hour per average man.

The chart demonstrates how quickly metabolic processes, as measured by gaseous exchange, reach a new equilibrium when the degree of physical activity

* [b] See bibliography.

changes. The very low mechanical efficiency of the human body as a machine for performing this type of work is demonstrated by the chart. The mechanical energy expended on the machine to raise the weight, section *D*, or to raise and lower it, section *E*, is indicated by *F* and *G*, respectively. The mechanical energy is expressed in calories per hour per square meter of body surface area for the average subject and in Btu per hour for a body surface equal to that of an average sized man.

Raising and lowering the weight *E* requires the transformation of 381 Btu per hour of energy within the body more than is required when the subject is standing still, while the useful work performed (*G*) is only 42.1 Btu per hour. Assuming that the 409 Btu (*B*) are useful for maintaining life while standing and 381 Btu are needed for raising and lowering the weight, the human body as an engine for performing this type of work is only 11.1 per cent efficient. A like consideration showed that for raising the weight only (*F*) the body is 19.8 per cent efficient. The last figure is in close agreement with the value of 16 per cent reported by Murlin[l] as the result of laboratory experiments with the same type of work. It is also in agreement with the value of 20 per cent given by Yagloglou[m], who plotted against the rate of work, the heat loss data, for men working, reported by Benedict and Cathcart[n] and Benedict and Carpenter[o]. On first thought it is surprising that more energy is required in raising the weight alone than is required for raising and lowering it. However, upon further consideration, the reason for this becomes apparent. While being lowered, the weight performs work in helping to raise the arms and body of the subject equal to its mass times the distance through which it falls, minus the work necessary to overcome friction in the system.

The relation between rate of working and energy production in the body, total heat loss, heat loss by evaporation, and heat loss by radiation and convection are given in curves *A*, *B*, *C* and *D*, respectively, Fig. 11. These data were all collected in a constant atmospheric condition of 70 deg effective temperature at 20 per cent relative humidity. The energy exchange at rest for these curves was taken from data published by the Research Laboratory[a-b] for persons at rest in the same air conditions. The three rates of work, 16,538, 33,075, 66,150 ft-lb per hour, were accomplished by raising and lowering the weight in the work machine at rates of once every 16, 8 and 4 seconds, respectively.

Curve *E* gives the rate at which mechanical energy was expended on the machine per square meter of the average body surface of the subjects and also for an average man. The relation between the metabolic rate or energy production and effective temperature for subjects raising and lowering the weight once every 8 seconds (working at the rate of 33,075 ft-lb per hour) is given in Fig. 12. The metabolic rate for individuals seated at rest is given in curve *D*.

The relation between effective temperature and total rate of energy and heat loss for subjects working at the rate of 33,075 ft-lb per hour is given in Fig. 13. The total rate of heat loss for subjects seated at rest and the rate at which mechanical energy was expended on the machine are given in curves *E* and *F*, respectively.

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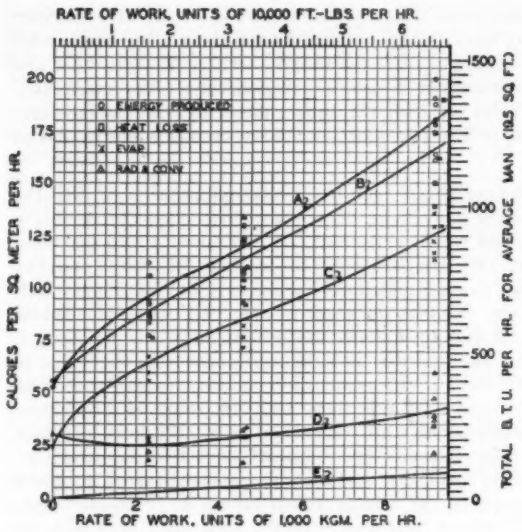

FIG. 11. RELATION BETWEEN ENERGY PRODUCTION AND HEAT LOSS FROM THE HUMAN BODY AND RATE OF WORK IN KILOGRAMS PER HOUR AND FOOT-POUNDS PER HOUR *

* Curve *A*—Energy production. Curve *B*—Total heat loss. Curve *C*—Heat loss by evaporation. Curve *D*—Heat loss by radiation and convection. Curve *E*—Mechanical energy expended on machine.

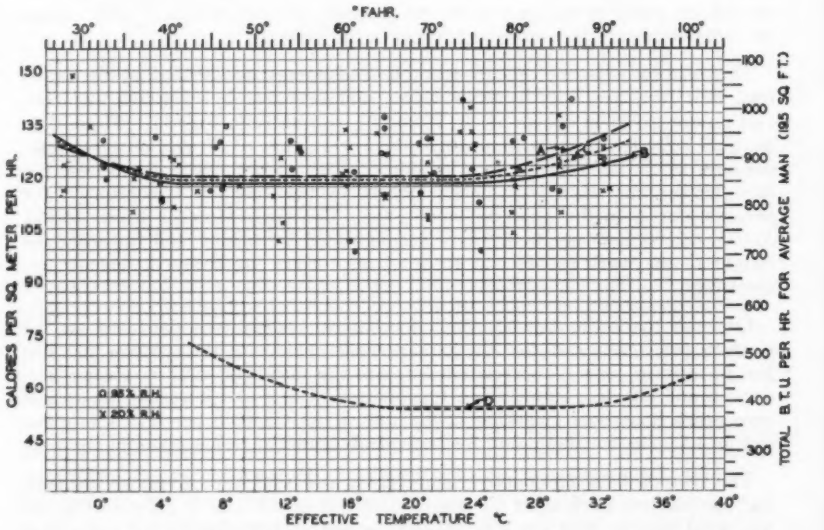

FIG. 12. RELATION BETWEEN ENERGY PRODUCTION IN THE HUMAN BODY AND EFFECTIVE TEMPERATURE *

* *A*—Energy production, at work, for 95 per cent relative humidity. *B*—Energy production, at work, for 20 per cent relative humidity. *C*—Average energy production, at work, all relative humidities. *D*—Average energy production, seated, at rest.

Latent and sensible heat losses from the entire body and from the respiratory tract are plotted against dry-bulb temperature in Figs. 14 and 15, respectively, for men working at the rate of 33,075 ft-lb per hour. The losses for men seated at rest are also given for comparison in the broken line curves. Heat and moisture losses from the respiratory tract of men working were calculated from the volume of normal air they would inhale and exhale for this rate of O_2 consumption, the temperature and moisture content of the air for each test

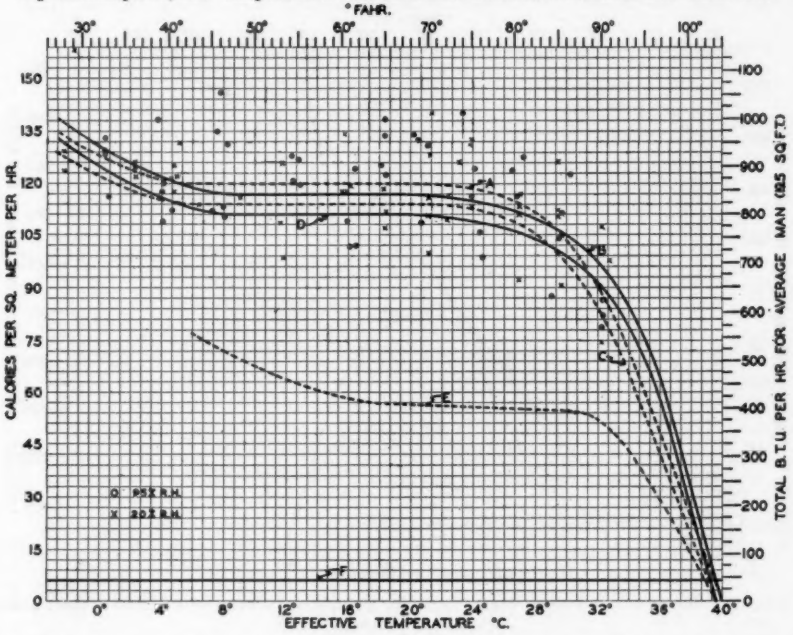

FIG. 13. RELATION BETWEEN ENERGY AND HEAT LOSS FROM THE HUMAN BODY AND EFFECTIVE TEMPERATURE *

* *A*—Energy loss, at work, for 95 per cent relative humidity. *B*—Energy loss, at work, for 20 per cent relative humidity. *C*—Heat loss, at work, for 95 per cent relative humidity. *D*—Heat loss, at work, for 20 per cent relative humidity. *E*—Average heat loss, seated, at rest. *F*—Mechanical energy expended on machine.

condition and the temperature and degree of saturation of exhaled air as found by experiment.[8] Respiratory losses are shown to be very small in comparison to other losses.

The relation between effective temperature at 95 per cent relative humidity and total heat loss, heat loss by radiation and convection and heat loss by evaporation are given in Fig. 16 for subjects working at the rate of 33,075 ft-lb per hour. In Fig. 17 similar data are given for air conditions of 20 per cent relative humidity. The broken line curves give the losses for men seated at rest for comparison.

[8] See bibliography.

The curves showing thermal reactions of men at work in Figs. 11 to 17 are similar in shape to the corresponding curves for subjects at rest, as previously published by the Laboratory[a-b] and shown in these charts for comparison. The magnitude of most of the reactions are, however, much greater for men working. Any characteristic changes in the slope in the working curves occur at lower temperatures than in the corresponding curves for rest.

A minimum energy production of 855 Btu per hour (119 cal per square

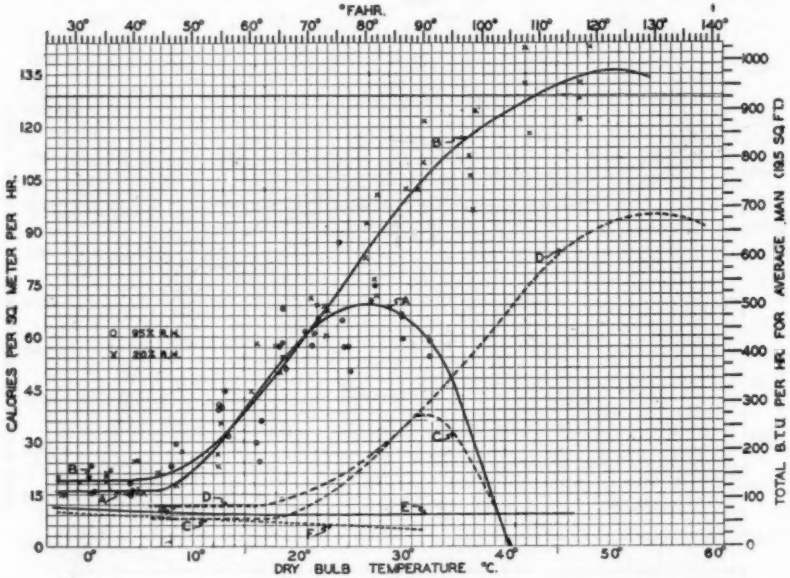

FIG. 14. RELATION BETWEEN HEAT LOSS BY EVAPORATION FROM THE HUMAN BODY AND DRY-BULB TEMPERATURE *

* *A*—From body surface and respiratory tract, at work, for 95 per cent relative humidity. *B*—From body surface and respiratory tract, at work, for 20 per cent relative humidity. *C*—From body surface and respiratory tract, seated at rest, for 95 per cent relative humidity. *D*—From body surface and respiratory tract, seated at rest, for 20 per cent relative humidity. *E*—From respiratory tract, at work, at 20 per cent relative humidity. *F*—From respiratory tract, at work, at 95 per cent relative humidity.

meter per hour) for an average man is indicated, which agrees with other investigators previously cited. Energy production and loss remain fairly constant over an effective temperature range of from about 43 to 75 deg and when plotted against effective temperature are practically independent of relative humidity, indicating that these reactions are functions of the effective temperature scale. If plotted against dry-bulb temperature, these reactions show considerable separating out for the two extreme humidities studied. Above an effective temperature of 75 deg, the total heat loss decreases rapidly until it reaches zero at about 103 F or at an effective temperature equal to 103 F body temperature. This decrease in heat loss, however, is not accom-

[a-b] See bibliography.

panied by a decrease in energy production but rather by an increase or in other words, the human body while working at this rate loses thermal control in an atmosphere above an effective temperature of 75 deg.

Fig. 11 shows that increased rate of work is accompanied by a rapid increase in energy transformation within the body many times greater than the rate

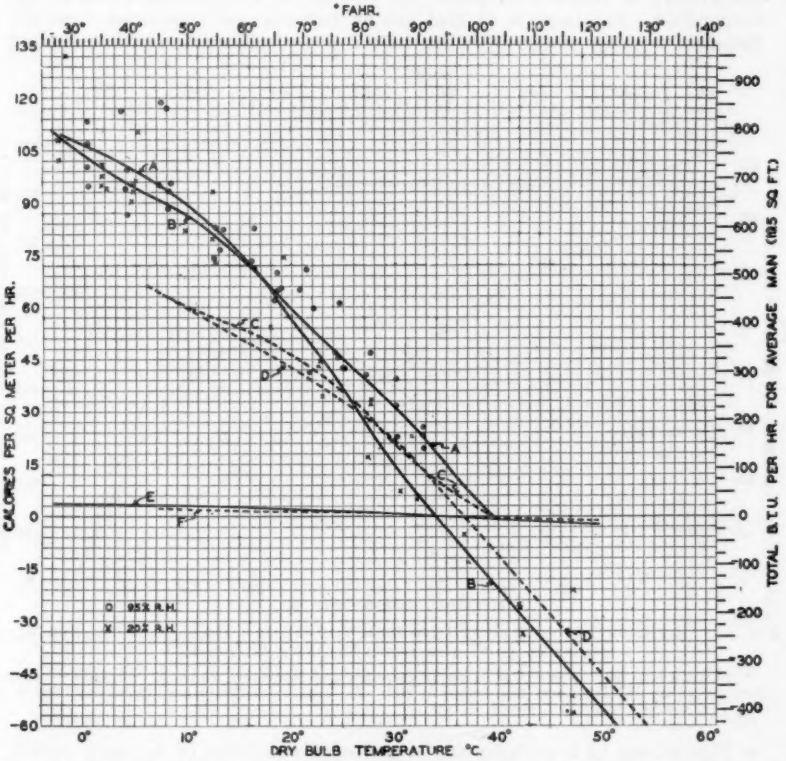

FIG. 15. RELATION BETWEEN HEAT LOSS BY RADIATION AND CONVECTION FROM THE HUMAN BODY AND DRY-BULB TEMPERATURE *

* *A*—From body surface and respiratory tract, at work for 95 per cent relative humidity. *B*—From body surface and respiratory tract, at work, 20 per cent relative humidity. *C*—From body surface and respiratory tract, seated at rest, for 95 per cent relative humidity. *D*—From body surface and respiratory tract, seated at rest, for 20 per cent relative humidity. *E*—Sensible heat loss from respiratory tract, at work, for 20 per cent and 95 per cent relative humidity. *F*—Sensible heat loss from the respiratory tract seated at rest, for 20 per cent and 95 per cent relative humidity.

of increase in the mechanical energy expended on the machine. Total heat loss and heat loss by evaporation also increase with work. Heat loss by radiation and convection, however, increases slowly. This is further borne out by the curves in Figs 14 to 17. At a dry-bulb temperature of 68 F, the heat loss by radiation and convection for men working is only about 33 per cent greater

than for men seated at rest. At the same temperature the heat loss by evaporation is 375 per cent greater for work than it is for resting conditions. This is in keeping with the physical conception of heat loss, for heat loss by radiation and convection should theoretically be independent of physiological control except for a slight variation in skin temperature. Physiological control of body temperature is obtained primarily by control of perspiration available for evaporation. The increase of heat loss by radiation and convection from the bodies of working subjects over the same loss for subjects at rest as indicated

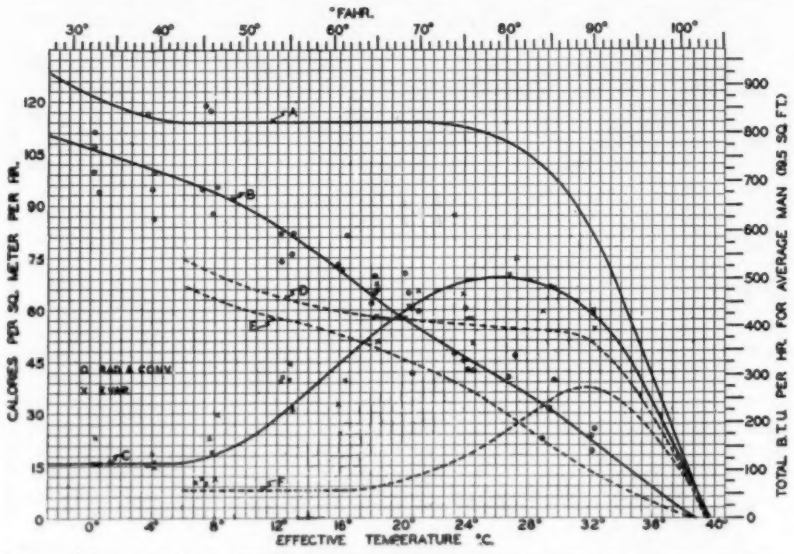

FIG. 16. RELATION BETWEEN HEAT LOSS FROM THE HUMAN BODY AND EFFECTIVE TEMPERATURE AT 95 PER CENT RELATIVE HUMIDITY *

* *A*—Total heat loss, at work. *B*—Loss by radiation and convection, at work. *C*—Loss by evaporation, at work. *D*—Total heat loss, seated at rest. *E*—Loss by radiation and convection, seated at rest. *F*—Loss by evaporation, seated at rest.

in Figs. 11 and 15 is probably due to increased ventilation of the body surface by the bellows effect of the clothes and the general air motion produced by the movement of the worker.

At dry-bulb temperatures of 103 F and 93.2 F, the heat loss by radiation and convection becomes zero for relative humidities of 95 and 20 per cent, respectively. The fact that heat loss by radiation and convection becomes zero at a lower temperature at 20 per cent relative humidity than it does at 95 per cent is probably accounted for by a lowered skin and clothing temperature at 20 per cent relative humidity due to evaporation of perspiration. The high temperature of 103 F, at which radiation and convection becomes zero at a relative humidity of 95 per cent, is probably fixed by the high body temperature and skin temperature attained by the subjects while working at this high temperature.

The composite charts (Figs. 16 and 17) illustrate the variation in thermal reactions with effective temperature at the same relative humidity. Total heat loss when plotted against effective temperature is independent of relative

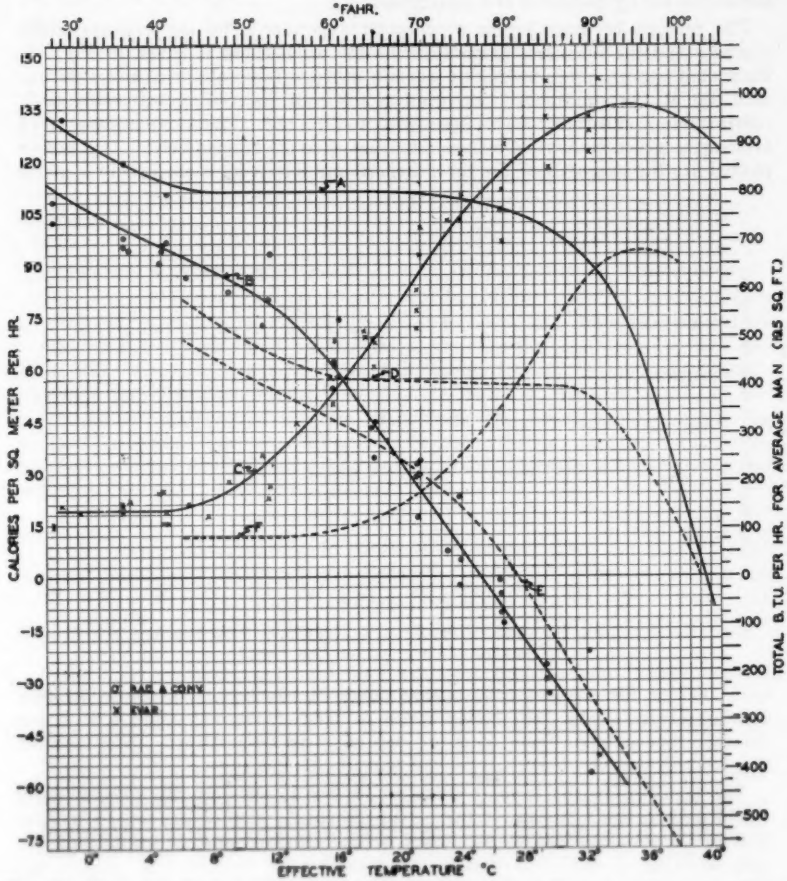

FIG. 17. RELATION BETWEEN HEAT LOSS FROM THE HUMAN BODY AND EFFECTIVE TEMPERATURE AT 20 PER CENT RELATIVE HUMIDITY *

* *A*—Total heat loss, at work. *B*—Loss by radiation and convection, at work. *C*—Loss by evaporation, at work. *D*—Total heat loss, seated at rest. *E*—Loss by radiation and convection, seated at rest. *F*—Loss by evaporation, seated at rest.

humidity. Radiation and convection and evaporation losses vary greatly with relative humidity for the same effective temperature as indicated by a comparison of the curves in the two charts. At an effective temperature of 61.5 deg and a relative humidity of 20 per cent, radiation and convection and evapo-

ration losses for subjects working are equal, each accounting for half of the entire heat loss, while at the same temperature at 95 per cent relative humidity, evaporation loss is only 39.7 per cent of the total, the other 60.3 per cent being accounted for by radiation and convection.

The relation between degree of comfort experienced by the subjects while working and the effective temperature of the atmosphere is given in Fig. 18. Each man expressed his feeling of warmth at equal intervals three times during each test at any atmospheric condition as too cold, too cool, comfortably cool, comfortable, comfortably warm, too warm and too hot. These expressions were arbitrarily given numerical values of from 1 to 7, respectively. The points plotted are the averages of the three expressions given by each man in a given

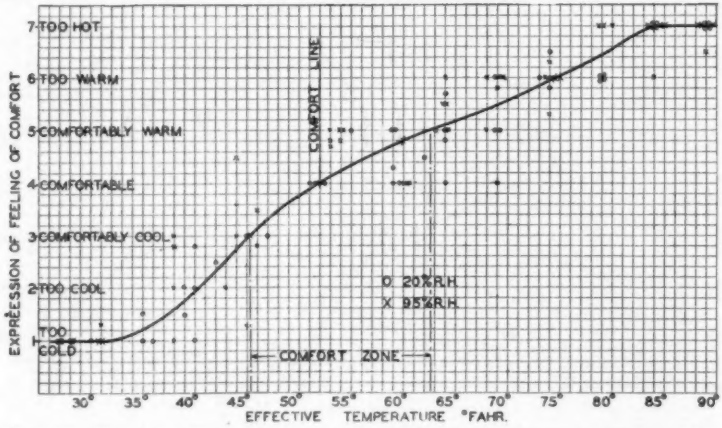

FIG. 18. RELATION BETWEEN COMFORT AND EFFECTIVE TEMPERATURE FOR MEN WORKING, 33,075 FT-LB PER HOUR

test. Taking the comfort zone as that range of effective temperature between comfortably cool and comfortably warm or from numerical value 3 to 5, the comfort zone extends from 46 to 64 deg effective temperature. The comfort line for this degree of work is 53 deg effective temperature.

The comfort line as located above is 11 deg lower than 64, as previously established by the Laboratory[p] for persons normally clothed at rest. The relation of the comfort line to the limits of the comfort zone as established for men working is similar to that reported earlier for men at rest, that is, the comfort line is nearer the lower limit of the zone than the upper limit. This means that a person is more sensitive to a few degrees of cooling below an ideally comfortable temperature than he is to the same degree of rise of temperature above. It should be emphasized that the comfort zone as established above is for men wearing coat and vest and working at the definite rate reported. In practice the coat and vest would be discarded when the man began to feel too warm and as a result he would be comfortable over a wider range of temperatures.

[p] See bibliography.

TABLE 2. CONDITIONS OF SENSIBLE PERSPIRATION FOR VARIOUS ATMOSPHERIC CONDITIONS

Degree of Perspiration *	Atmospheric Conditions											
	Men Working 33075 Ft.-Lb. per Hour						Men Seated at Rest					
	95 Per Cent Relative Humidity			20 Per Cent Relative Humidity			95 Per Cent Relative Humidity			20 Per Cent Relative Humidity		
	E.T.	D.B.	W.B.	E.T.	D.B.	W.B.	E.T.	D.B.	W.B.	E.T.	D.B.	W.B.
Forehead clammy	59.0	59.4	58.3	69.5	80.5	56.5	73.0	73.6	72.4	75.0	87.0	60.7
Body clammy	50.0	50.2	49.3	57.0	61.6	44.2	73.0	73.6	72.4	75.0	87.0	60.7
Body damp	60.0	60.3	59.3	62.5	69.6	49.5	79.0	79.7	78.4	81.0	97.5	67.5
Beads on forehead	68.0	68.5	67.5	76.0	91.0	63.4	80.0	80.8	79.4	87.0	109.4	75.2
Body wet	69.0	69.6	68.5	71.0	82.8	53.0	84.5	85.4	84.0	86.5	108.5	74.6
Perspiration on forehead runs and drips.....	78.5	79.3	78.0	82.0	100.5	70.2	88.0	89.0	87.6	94.0	125.2	85.4
Perspiration runs down body	79.0	79.8	78.5	81.0	99.8	69.0	88.5	89.5	88.1	90.0	116.0	79.5

* 40 per cent of subjects registered degree of perspiration equal or greater than indicated.

The effective temperature at which perspiration changes from insensible to sensible for men working is lower than for men at rest. This is indicated by the heat-loss-by-evaporation curves for persons at rest and working in Figs. 16 and 17. The atmospheric conditions at which various degrees of perspiration take place for men working are given in Table 2. The lowest degree for which 40 per cent of the subjects experienced the condition of perspiration indicated is given for both 20 and 95 per cent relative humidity. For comparison the conditions at which the same degree of perspiration takes place for persons at rest are also given.

SUMMARY

1. This report contains data in the form of curves giving the rate of total heat loss, heat loss by radiation and convection and heat loss by evaporation for men working at three constant rates in still air at various temperatures and two humidities. Corresponding curves for these losses with subjects seated at rest are also given.

2. Data for practical application and examples showing how it may be used in the solution of problems in air conditioning are presented.

3. A table showing the metabolic rate for different kinds of activity is given and the application to practical problems is shown.

4. Total energy production, energy loss and heat loss are shown to be functions of effective temperature.

5. Sensible and latent heat loss are shown to be functions of dry-bulb temperature and only slightly affected by relative humidity except at extreme temperatures.

6. Sensible heat loss for men working increases but little over this loss for men at rest.

7. Latent heat loss increases rapidly with physical activity and is depended upon almost solely by the body for maintaining a constant body temperature with varying rates of heat production.

8. The comfort zone for men normally clothed and working at 33,075 ft-lb per hour is given as 46 to 64 deg effective temperature and the comfort line as 53 deg.

9. The degree of perspiration experienced by men working at 33,075 ft-lb per hour is given for various temperatures and humidities.

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DISCUSSION

W. L. FLEISHER (WRITTEN): The following important facts in connection with this paper can be expanded to more thoroughly cover the data required for the solution of the air conditioning problem.

It is obvious that in dealing with heat released into the atmosphere by the metabolism of people either at rest or at work, in most cases to relieve the condition brought about by this congestion of people or activity in an enclosure, the air conditioning engineer would be confronted with the refrigeration requirements of such a problem.

In dealing with such a problem the following three points should be considered.

In the first place, the rise in sensible temperature, due to the metabolism of the occupants of the room, would have to be determined so that sufficient air could be brought in at a temperature low enough to maintain a pre-determined dry-bulb condition irrespective of the number of occupants in the room or enclosure. This would

seem to present few difficulties because the authors have obtained the essential data required. However, in all problems of this kind stress should be given to the cooling effect due to the absorption by the air of the moisture being given off by the occupants. This cooling effect is due to the thermodynamic relationship that the air must give off sufficient heat in order to evaporate the water into a gas or vapor. Approximately 1,000 Btu are deductible from the sensible heat gain for each pound of water evaporated into the atmosphere from the room or its occupants. Now, if this moisture is thrown off from the people in the form of a gas, transformation has already taken place, and the cooling due to a change of state would not exist. Consequently, a mistake in the calculations if developed as in the past would undoubtedly occur.

The evaporation of the moisture given off as perspiration would create a cooling effect which would be to the advantage of the dry-bulb temperature, as far as cooling is concerned. Therefore, it is essential that the amount of moisture given off by the respiratory system and the amount given off by perspiration should be differentiated so that these quantities can be correctly applied in the equation.

The refrigeration problem is concerned with the total heat of the air entering the washer and being maintained in the enclosure, and the total heat of the wet-bulb. This is the sum of the sensible heat above zero and the latent heat above zero of the moisture which must be brought down to a new dew-point sufficiently low so that it can take care of the sensible heat from the people. It also must contain sufficiently few grains of moisture so that when it has absorbed the moisture given off by the people it will still have a dew-point low enough to maintain the comfort condition specified.

As a definite condition of both temperature and relative humidity is usually called for when apparatus is used, the work of the air conditioning engineer has created problems which call for a definite variable control. This work, more than any other research, indicates that the method of hand control is inadequate to take care of the changing metabolism which takes place so markedly in human beings under different states of rest or work.

If 400 Btu is taken as the total energy loss between effective temperature of 60 F and effective temperature of 90 F, and the sensible heat loss between temperatures of 70 and 90 F dry-bulb are taken as between 350 and 100 F, the balance of the heat losses would be latent heat losses; in other words, from 300 to 50 F. Between these same temperatures the total grains of moisture vary from approximately 700 to 1,800 per person per hour. This gives an interesting table for the refrigeration requirements for maintaining equal comfort conditions. In fact, the data obtainable indicates that equal comfort conditions could not be maintained. For instance, with 75 F dry-bulb and 60 per cent relative humidity, which is a common condition to be maintained in an auditorium, we would have a dew-point of 60 F. Under these conditions a human body at rest would give off 850 grains of moisture per hour, or 14 grains of moisture per minute. With 30 per cent of the air going through the washer or 9 cu ft per person per minute, 1⅔ grains of moisture would have to be carried in this air below the dew-point required to maintain the condition in the room.

At 60 F dew-point where we have 5.7 grains per cubic foot, we would have to reduce to approximately 4 grains per cubic foot or a dew-point of 49. Or if we took 40 per cent through the washer or 12 cu ft we would have to reduce to approximately 54 F dew-point to absorb the moisture given off by the people and still maintain the relative humidity of 60 per cent that we desire.

However, if we were going to maintain equal comfort in the theater at 80 F, according to the comfort chart we could not have over 30 per cent relative humidity. This gives us a 45-F dew-point but at 80 F we have 1,200 grains per person per hour which is 20 grains per person per minute to be absorbed in, we will say, 9 cu ft

or 2⅞0 grains per cubic foot, a 19-F dew-point which is an absolutely impossible condition. In the first place, without brine such a condition could not take place, and when it is considered that the outside air at 75 F wet-bulb would have to be brought down to this condition, the refrigeration requirements would be beyond all reason.

Obviously, something is wrong either in the assumptions of summer comfort or in the relationship between sensible and latent heat and grains of moisture. Usually, 80 F and 50 per cent relative humidity, which gives approximately the same dew-point as 75 F and 60 per cent relative humidity, has been considered approximately similar conditions for theater comfort. Both of these conditions have the same dew-point of 60 F and consequently the same number of grains of moisture to be maintained. However, according to the new charts, the grains of moisture given off at an 80-F dry-bulb which must be absorbed in the surrounding air are considerably more than at 75 F and consequently the refrigeration requirements with air at a

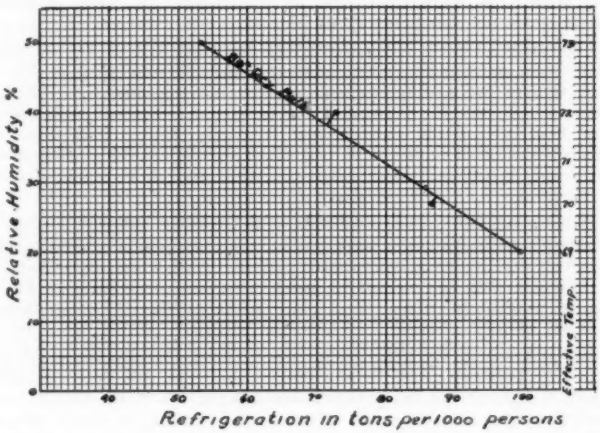

FIG. A. EFFECTIVE CONDITIONS FOR EQUAL AND VARYING REFRIGERATION FOR EQUAL EFFECTIVE LINES

definite wet-bulb being brought down to a lower dew-point would obviously require more refrigeration for a higher indoor temperature.

As the refrigeration requirements are a measure of the total heat to be taken care of, possibly an equal refrigeration curve would give equal conditions as far as comfort is concerned. Fig. A shows the effective conditions for equal refrigeration and varying refrigeration for equal effective lines.

It is also interesting to note that a number of tests taken over most of one season in a moving picture house, in which the number of people were plotted against the total heats by means of a measurement of entering dry- and wet-bulb as against leaving dry- and wet-bulb, show 700 grains of moisture per person per hour which was almost uniformly obtained with conditions of 77 F dry-bulb, 68 F wet-bulb, and a relative humidity of 55 per cent.

DR. W. J. MCCONNELL (WRITTEN): The authors have presented a most interesting and vivid description of human physiological responses to varying degrees of work from the viewpoint of heat and moisture losses, and have indicated to the engineer many practical applications of the results of these studies.

Notwithstanding the fact that the influence of temperature upon various physiological activities has been extensively studied, unfortunately, comparatively few such experiments have been made under controlled conditions. While Rubner, as early as 1896 estimated that a man comfortably clothed and sedentary in a room at 15 C, with the air 50 per cent saturated with moisture, loses 44 per cent of his total heat dissipation by radiation, 30 per cent by convection, and 20 per cent by evaporation from lungs and skin, and Aldrich in 1928 arrived at much the same conclusions, we know that the proportions are different with changes in surrounding temperature and with activity. The paper here presented gives us accurate information with definite measurements of work correlated with various degrees of effective temperature.

The many practical applications suggested by the authors and charts indicating the relationship existing between the various factors studied enhance the value of the article which is an important contribution to scientific medicine and the art of air conditioning.

PROF. A. C. WILLARD (WRITTEN): This paper is a most valuable contribution to our knowledge of the amount of the heat and moisture losses from men engaged in various degrees of physical activity, and the relative amount of such losses over wide ranges of temperature.

The effect of the physical activity of a person on the heat which must be lost from the human body and the further effect of the temperature of the surrounding air on these losses is not generally appreciated even by heating and ventilating engineers. We are so accustomed to allowing approximately 400 Btu per hour for the total heat loss from the average adult when at rest, as in an auditorium with the surrounding air at about 70 F, that this value has become more or less fixed in our minds and is often used incorrectly. In some cases, no account is made of the fact that roughly ¾ of this heat is given off as sensible heat by increasing the temperature of the surrounding air, while the remaining ¼ is given off as latent heat of evaporation and is contained in the vapor of the moisture evaporation from the body. This heat in the vapor is of no consequence until we attempt to cool the air below its dew-point, at which time it must be removed in order to condense the vapor and hence becomes a troublesome factor in solving the average air cooling problem.

One of the most interesting applications of the data in this paper is to be found in determining the probable rise in temperature of the air in a subway station at the time of the afternoon rush period. As the authors have indicated in the sample problem, No. 3, it is necessary not only to know the number of people involved but also to estimate the probable average activity of a subway crowd, which in the example in question is assumed to be equivalent to walking at the rate of about 2 mph. To anyone who is not familiar with this problem it may be surprising to find that the Btu loss per hour for the average adult when walking 2 mph is about 760, which is nearly twice as much as for the same adult seated at rest.

The problem becomes still further complicated when we take into consideration the unusual temperature and humidity conditions which may exist in the subway station during the rush hours of the summer months, at which time the dry-bulb temperatures may go above 90 F and the relative humidity becomes largely a matter of speculation. Without such data as are presented in Figs. 1, 2 and 3 of this paper, no rational solution of the problem is possible. However, the authors have clearly shown how both the sensible and latent heat losses from the average adult may be estimated under the atmospheric conditions existing in a subway station, and the principal purpose of this discussion is to point out the fact that when a man is walking at the rate of 2 mph with the surrounding air at 80 F dry-bulb and at 66 F wet-bulb, he loses about 240 Btu per hour in sensible heat and about 530 Btu per hour in latent heat. The same man seated at rest with the air at 70 F would lose

about 300 Btu per hour in sensible heat and only 100 Btu per hour in latent heat. Finally, in the former case he adds water vapor to the surrounding air at the rate of 3,580 grains per hour while in the latter case he will add only about 750 grains of water vapor per hour. Thus, is strikingly brought out the fact that as the surrounding air temperature rises, a man loses less and less sensible heat and a rapidly increasing amount of latent heat which is contained in the water vapor given off by the evaporation of the body moisture.

The almost unlimited application of the principles set forth in this paper to the various air conditioning or ventilating problems of both public and private enterprises cannot be over estimated and both the authors and the Society are to be congratulated in presenting for record such complete and valuable data.

F. C. HOUGHTEN: The discussion of the application of the laboratory data to problems in theater cooling by Mr. Fleisher is most valuable as a means of correlation between these data collected under definitely specified laboratory and practical conditions.

It should be emphasized that the total heat dissipated from the body of an average size man at rest, or about 400 Btu for normal temperatures, consists of both sensible and latent heat. The latent heat loss whether resulting from evaporation in the respiratory tract or from the body surface does not raise the dry-bulb temperature of the air.

In maintaining a constant atmospheric condition in the presence of people, the air conditioning system must remove both sensible heat and moisture. Since dehumidification is usually accomplished by lowering the temperature to a point below the prevailing dew-point, the removal of moisture is also accompanied by removal of sensible heat. If it is satisfactory to maintain such an atmospheric condition that the cooling necessary to remove the moisture added by the people present also is effective in removing just the required amount of sensible heat, then that will be the most efficient cooling cycle.

It has been shown that the same degree of comfort as regards one's feeling of warmth is had for combinations of dry-bulb temperature and moisture content giving the same effective temperature. While we have no conclusive proof that certain relative humidities are more conducive to health and comfort than others, it is generally assumed that the humidity should be kept within the range of 30 to 70 per cent. This would allow considerable variation in the dry-bulb temperature and moisture content of the air for a given degree of comfort.

The higher the dry-bulb temperature of the atmosphere, the greater the percentage of latent over sensible heat which must be removed in order to maintain constant atmospheric conditions. Conversely, for the same effective temperature, the higher the moisture content of the air, the greater the percentage of sensible and the lower the percentage of latent heat dissipated.

Thus, for the condition of 80 F dry-bulb and 30 per cent relative humidity cited by Mr. Fleisher, 215 Btu sensible and 175 Btu latent heat must be removed per 150-lb man in order to maintain constant air conditions. In order to remove 175 Btu latent heat and 1,200 grains of moisture per hour with the 9 cu ft of air per minute specified or 39.1 lb of dry air per hour, 30.7 grains of moisture and 4.5 Btu latent heat must be removed per pound of dry air handled. In order to accomplish this, the air would have to be cooled to a dew-point of approximately 20 F. The total heat above zero contained in 1 lb of dry air at 80 F dry-bulb, 30 per cent relative humidity and 60.2 F wet-bulb is 26.2 Btu. The total heat above zero contained in 1 lb of dry air at 20 F dry-bulb and 100 per cent relative humidity is 7.0 Btu. Hence, the removal of the 4.5 Btu latent heat is accompanied by a total heat removal of 19.2 Btu. Not only is the low dew-point temperature of 20 F difficult, if not impossible, to obtain practically, but the cooling cycle is an inefficient one for the

reason that in order to remove the 30.7 grains of moisture, or 4.5 Btu latent heat per pound of dry air, it is necessary also to remove 14.7 Btu sensible heat per pound of dry air while the removal of 5.5 Btu is all that is required, assuming no heat gain from other sources.

The condition of 80 F dry-bulb and 30 per cent relative humidity maintained gives 71.8 F effective temperature. If this same effective temperature were maintained with 70 per cent relative humidity, the dry-bulb temperature would be 75 F, and the sensible and latent heat dissipated per 150-lb man would be 262 and 137 Btu per hour, respectively.

In order to remove the 137 Btu latent heat or 920 grains of moisture per hour with 9 cu ft of air per minute or 39.4 lb of dry air per hour, 3.48 Btu of latent heat or 22.8 grains of moisture must be removed per pound of dry air handled. The moisture content and total heat above zero of 75 F air at 70 per cent relative humidity are respectively 90.4 grains and 31.7 Btu per pound of dry air. In order to remove the 22.8 grains moisture per pound of dry air, the moisture content will have to be reduced to 67.6 grains per pound of dry air, which will be accomplished by cooling to a dew-point of 56.3 F where the air will have a total heat above zero of 23.8 Btu. In order to remove the 22.8 grains of moisture and the 3.48 Btu latent heat, it was necessary to remove only 4.42 Btu sensible heat per pound of dry air, which is less than the 6.65 Btu actually added per the 150-lb man.

Thus, for the lower dry-bulb and the assumed effective temperature, the removal of the moisture added by one 150-lb man required the removal of only 7.9 Btu total heat, while for the same effective temperature and 30 per cent relative humidity the removal of the moisture added by the same person required the removal of 19.2 Btu total heat.

It is true that for maintaining the higher dry-bulb much more heat was removed than actually added by the person, and assuming no heat gain by the atmosphere from other sources, heat would have to be added in order to maintain constant conditions. In the case of the 75 F dry-bulb temperature, the removal of the 7.9 Btu total heat was not sufficient to take care of both sensible and latent heat dissipation of the 150-lb man, and additional cooling would have to be resorted to. It is obvious that for some intermediate dry-bulb temperature giving the same effective temperature of 71.8 F the necessary cooling for dehumidification would just serve to remove the right amount of heat to take care of both sensible and latent heat dissipation, assuming perfection in accomplishing the application of cooling to the atmosphere and no heat exchange between the air conditioned space and the surrounding other than the heat dissipation from the persons present.

Of necessity the data were collected in the laboratory for definite conditions of rest and activity of the subjects. For the condition of rest, the activity of the subjects was made to simulate as closely the activity of persons in a school or theater as was consistent with the actual requirements of the test. The subjects were seated at a table with freedom to move their bodies as desired while in the seated position. The conditions imposed by the test procedure made it necessary for the subjects to occasionally leave the seated posture for weighing and sampling of exhaled breath.

Metabolism, total heat loss, and heat loss by radiation and convection and by evaporation have been found to be functions of the body surface area of the subjects. Hence, the size, including weight and height of a person must be taken into consideration in evaluating these thermal exchanges. The laboratory data were collected on men of average size and are plotted both per unit of body surface area and against a surface area of 19.5 sq ft, which is the surface area of an average size man 5 ft 8 in. in height weighing 150 lb.

While clothing does not materially affect the total rate of heat dissipation from the body, it is found to affect the relative proportion of heat dissipation as sensible and latent heat. The greater the amount of clothing worn or the better the insulation

of the human body, the lower the rate of heat dissipation from the surface in the form of sensible heat. Therefore, in order to maintain temperature equilibrium within the body, heat elimination in the form of latent heat is increased with increased weight of clothing.

In applying laboratory data to theater audiences, all these factors must be taken into consideration. Thus, while the laboratory curves show approximately 1,000 grains of moisture elimination from the body of a man 5 ft 8 in. in height and weighing 150 lb in an atmosphere of 77 F dry-bulb temperature, the latent heat dissipation from an average theater patron would be considerably less. Considering that a large proportion of women are in such an audience together with a smaller proportion of the younger generation, an average body surface area of 16.7 sq ft representing a person 5 ft 3 in. in height and weighing 120 lb would more nearly represent average conditions. The laboratory data would give approximately 850 grains of moisture evaporation from such a person. This value is still high compared with the 700 grains for the same temperature found in an actual theater by Mr. Fleisher. Other factors, however, probably account for the discrepancy, such as the lighter clothing worn by the average theater patron, and possibly a lower degree of physical activity than that applying to the subjects participating in the laboratory tests.

The questions brought up in Mr. Fleisher's paper serve to emphasize that care and judgment must be exercised in the application of the laboratory data to any particular condition of human activity in a given atmospheric environment. It should be strongly emphasized that the laboratory data represents a true condition of thermal exchanges between the human body and the atmospheric environment for the degree of activity, type of subject, clothing and other environmental conditions prevailing in the tests. However, the laboratory data for the condition of rest studied and the three degrees of physical activity makes it possible to interpolate between these results for other degrees of activity, provided the particular type of activity in question and the resulting metabolic rate is understood.

Further, the facts brought out in the paper, that these thermal exchanges are functions of body surface area, and that increased clothing results in increased latent and decreased sensible heat loss, makes it possible to apply the data to actual conditions found in audience halls and workshops to at least as high a degree of accuracy as the environmental conditions in question can be ascertained. Hence, the value in such data as presented by Mr. Fleisher in making possible the correlation of the accurate findings of the laboratory for a few special and accurately controlled environmental conditions to the varied conditions found in practice. A vast amount of such practical information gleaned from practice in the field, and analyzed in the light of the laboratory findings will be necessary in order to thoroughly and sufficiently understand the practical environmental conditions so that the laboratory data may be applied to them with a high degree of accuracy.

No. 909

AIR INFILTRATION THROUGH DOUBLE-HUNG WOOD WINDOWS

By G. L. LARSON [1] *(MEMBER)*, D. W. NELSON [2] *(MEMBER)*, AND R. W. KUBASTA [3] *(NON-MEMBER)*, MADISON, WIS.

This paper is the result of research conducted at the University of Wisconsin in cooperation with the A. S. H. V. E. Research Laboratory.

AIR infiltration through various types of building constructions has been the subject of research carried on at the University of Wisconsin for the past several years in cooperation with the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS. The previously reported work has been on walls of brick and wood frame constructions.[4] The primary purpose of the investigation reported in the present paper was to aid in the establishment of figures for both plain and weather-stripped windows of the double-hung wood type for use in the calculation of heat losses from buildings. As secondary aims, it was desired to study the variation to be expected from one window to another for various cracks and clearances, for locked and not-locked conditions, and for several representative weatherstrips as applied to plain windows showing both low and high resistance to air leakage.

PROGRAM AND WINDOW USED

For the investigation, fourteen windows having a sash opening of 3 ft 0 in. by 6 ft 0 in. were purchased. The thickness of the sash was 1⅜ in. in all cases, and for convenience in testing, box frames were used. These differed from standard masonry frames in the additional use of box construction at the head and sill and the use of steel corner plates to stiffen the frames for test purposes. Some of these windows are shown in Figs. 1 and 2.

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[4] See Air Infiltration through Various Types of Brick Wall Construction by G. L. Larson, C. Braatz and D. W. Nelson (A. S. H. V. E. TRANSACTIONS, Vol. 35, 1929) and Air Infiltration through Various Types of Wood Frame Construction by G. L. Larson, D. W. Nelson and C. Braatz (A. S. H. V. E. TRANSACTIONS, Vol. 36, 1930).

Presented at the Semi-annual Meeting of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, Swampscott, Mass., June, 1931, by D. W. Nelson.

Five of the windows were purchased a year in advance of the actual time of starting the test, which was in the late Summer of 1930, and from a mill from which windows of poor workmanship and material had previously been secured. These were intentionally specified to be of a poor grade. The other nine of the 14 were ordered from a mill from which windows of a good grade had been previously secured and were specified to be of the best material and workmanship throughout. These were delivered 3 months before the testing started.

FIG. 1. ARRANGEMENT OF TEST EQUIPMENT

From the time of delivery, all windows were kept in the laboratory to become seasoned before adjusting cracks and clearances or testing.

Six months after starting the tests, the 14 windows were weatherstripped by local representatives of three weatherstripping manufacturers. Each representative applied his weatherstrip to three of the windows termed to be of a good grade, and to two of the windows considered to be of a poor grade except that one make of weatherstrip was applied to only one window of poor grade since there were only 5 of this grade. For the purposes of this paper, the windows were designated by letters that follow in sequence for each of the three weatherstrips represented in the tests. Table 1 explains these designations and the symbols that were used on the curve sheets included in this paper.

TEST EQUIPMENT USED

Fig. 1 shows a general view of the test equipment arranged for window testing. The method of clamping a window in place is shown in Fig. 2. The

partition to which the window is clamped divides the machine into a pressure chamber and a collecting chamber. The leakage through the partition is maintained at a low value and the joint between the plane outside face of the window frame and the partition is sealed by the compression of a sponge rubber gasket strip. Distance blocks are located at each clamping location to insure the even

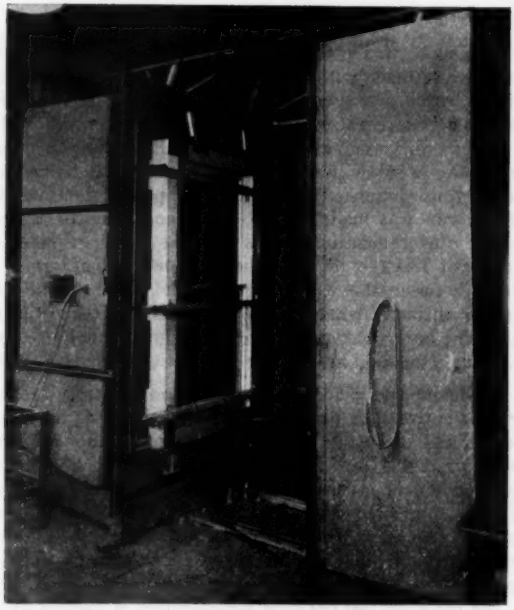

FIG. 2. WINDOW IN POSITION FOR TEST

compression on the gasket and the maintaining of the window in a plane undistorted condition.

Artificial wind pressure is exerted by a small motor-driven blower against the outside of the window which forms one part of the enclosure for the pressure chamber. The pressure drop through the window as indicated by an inclined draft gage connected to the two chambers is adjusted by means of dampers on the inlet and discharge sides of the blower. Air that passes through the partition wall and window into the collecting chamber is measured as it

TABLE 1. DESIGNATION OF WINDOWS AND WEATHERSTRIPS

WINDOW GRADE	GOOD			POOR	GOOD			POOR		GOOD			POOR	
WINDOW	A	B	C	D	E	F	G	H	I	J	K	L	M	N
CURVE SYMBOL	◓	◒	◐	◑	○	⊖	⊠	⊕	⊕	□	⊟	⊠	⊞	⊡
WEATHERSTRIP	← 1 →				← 2 →					← 3 →				

escapes into the atmosphere by thin-plate, square-edged orifices. The drop in pressure through the orifice is measured on a Wahlen gage. Access doors are provided in each of the two chambers so as to be able to open, close, lock or seal the window during the progress of a test without disturbing the rubber seal between the two halves of the machine.

TEST PROCEDURE

Each time a window was placed in the machine for testing, 4 runs were made. Two runs were made with the window closed but not locked. This was done to obtain a better average value than from one test. Before each of these tests, both sash were raised and lowered several times and the closure made with ordinary effort as it would be in actual building occupancy. Access to the inside of the window was secured through the bolted and gasketed door in the collecting chamber. Since locking places the sash in a definite position, only one run was made at this condition. The air leakage observed at the orifice in the aforementioned runs was the sash perimeter leakage plus a certain amount of leakage through the window frame joints, through the rubber seal joint between the window frame and the machine partition, and leakage occurring through whatever minute cracks existed in the machine partition itself. The latter leakage was kept at a low value by sealing all visible cracks with calking compound and by the application of asphalt paint.

Each time a window was placed in the machine for test, a fourth run was made in which the entire sash perimeter was sealed on the outside of the window to prevent all leakage of air at the joints between the sash and the frame at the head, sides or jambs, meeting rail and sill. The leakage obtained in this run was the leakage entering the window at the frame joints, and at the rubber seal, and that passing through the partition itself. This leakage was subtracted from the total leakage in the locked and not-locked runs to secure the net leakage occurring through the sash perimeter. The leakage in this sealed run is the leakage of the test set-up obtaining when a perfect weatherstrip is applied, hence the name of 100 per cent weatherstrip is applied to this run. In all of the tables and curves of this paper, this sealed run leakage has been subtracted except in the runs shown in Fig. 4 which illustrates methods of sealing. It should be noted that only a part of this sealed run leakage is window frame leakage and that there are no values for frame leakage obtainable from a study of the 14 frames built for the testing of sash perimeter leakage. Later in this paper, frame leakage is discussed as determined on frames built into wood frame and masonry wall sections.

ONE HUNDRED PER CENT WEATHERSTRIP

The purpose of a window is to admit light, to allow the entrance of air when it is wanted, and at other times to prevent all air leakage or infiltration. When the joints between the sash and the frame permit no entrance of air, the cash perimeter is perfect in so far as infiltration is concerned. This condition is secured by the sealing of the sash perimeter on the outside of the window.

To approach this condition in actual building construction, weatherstripping is generally applied. There are two major factors in weatherstripping that influence the effectiveness of the installation. One of these is the ability of

the strip to stop all air that reaches it and the other is the placing of the strip with respect to the sash perimeter joint so that it is in a position to stop all leakage entering this joint. Fig. 3 at *c* and *d* shows the two extreme positions for the placing of the strip. When placed as at *c* on the upper sash, it is possible for air that enters the sash perimeter joint to enter the upper sash pulley holes. When once in the weight spaces, access to the room is gained through the lower sash pulley holes and through cracks in the frame such as at the sash weight doors or around the window trim. Were the strip and its

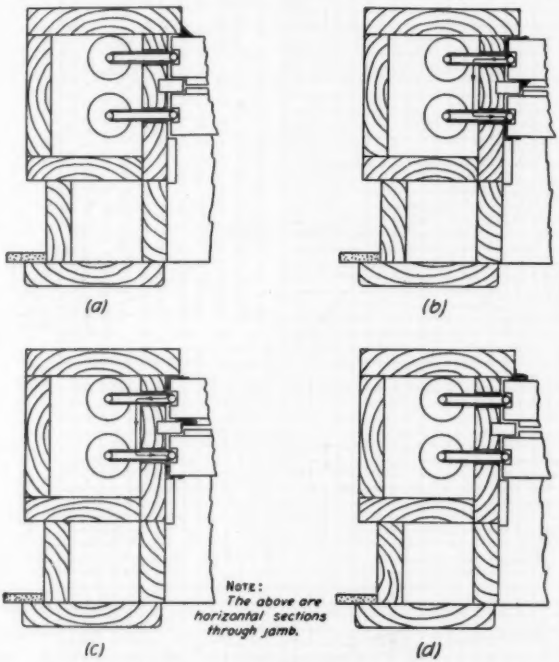

FIG. 3. TWO METHODS OF SEALING SASH PERIMETERS

location perfect, it would prevent the entrance of air at the outside contact line between the sash and the frame; the fact that it does not, means that it falls short of being perfect in tightness or in location.

Fig. 3 at *a* and *b* shows the two methods of sealing the sash perimeter to determine the leakage other than sash perimeter leakage. When the seal is applied to the inside of the upper sash as at *b*, it is still possible for air to enter at the sash perimeter and enter the room through the pulley holes and other smaller openings communicating with the sash weight spaces. It is conceivable that an actual weatherstrip applied to the outside of the upper sash

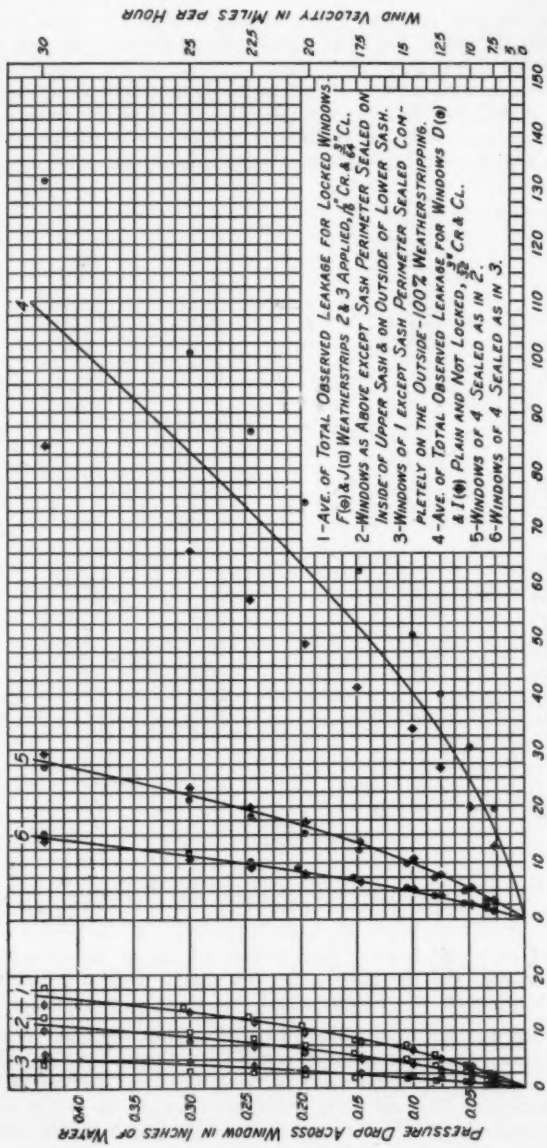
WIND VELOCITY IN MILES PER HOUR
PRESSURE DROP ACROSS WINDOW IN INCHES OF WATER
TOTAL INFILTRATION IN C.F.M.
1-AVE. OF TOTAL OBSERVED LEAKAGE FOR LOCKED WINDOWS F(a) & J(a) WEATHERSTRIPS 2 & 3 APPLIED, 1/8" CR. & 3/64" CL.
2-WINDOWS AS ABOVE EXCEPT SASH PERIMETER SEALED ON INSIDE OF UPPER SASH & ON OUTSIDE OF LOWER SASH.
3-WINDOWS OF 1 EXCEPT SASH PERIMETER SEALED COMPLETELY ON THE OUTSIDE-100% WEATHERSTRIPPING.
4-AVE. OF TOTAL OBSERVED LEAKAGE FOR WINDOWS D(a) & I(a) PLAIN AND NOT LOCKED, 3/32" CR & CL.
5-WINDOWS OF 4 SEALED AS IN 2.
6-WINDOWS OF 4 SEALED AS IN 3.

FIG. 4. COMPARISON OF RESULTS FOR TWO METHODS OF SEALING SASH PERIMETERS

as at *d* would stop more leakage than the perfect seal placed on the inner side where it could not prevent air reaching the pulley holes. The actual weatherstrip would then be more perfect than the seal set up as 100 per cent perfect. This obviously indicates a mis-location of the seal and points to the propriety of placing the seal on the outside of the upper sash as well as on the outside of the lower sash.

Fig. 4 shows a comparison of results using the two methods of sealing two plain windows having a high leakage and two weatherstripped windows having a low leakage. The difference between curves *1* and *2* is the leakage obtained for the weatherstripped window perimeter if the seal on the inner side of the upper sash represented 100 per cent weatherstripping. Likewise, the difference between curves *4* and *5* is the leakage through the sash perimeter for the plain window under the same conditions. The differences between curves *2* and *3*

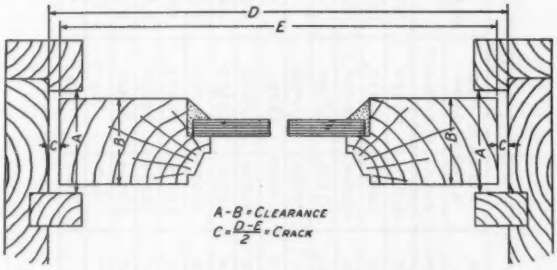

FIG. 5. DIAGRAM ILLUSTRATING CRACK AND CLEARANCE

and between *5* and *6* represent the leakages that originated in the sash perimeter and found a way into the room through the pulley holes and other smaller openings. This properly should be charged against the sash perimeter whether plain or stripped. It is included as sash perimeter leakage when the seal is made on the outside of both sash. This method of sealing was used in all tests reported in this paper except in the case of two illustrated in Fig. 4 to show the difference in the two methods. In comparing window tests, it is important to know which method of sealing was used. Fig. 4 shows that the leakage charged against the weatherstrip is only about one-half of that entering the sash perimeter if the seal is applied to the inside of the upper sash in the case of the two weatherstripped windows represented in this figure.

SETTING OF CRACK AND CLEARANCE

The 5 windows of a poor grade *D, H, I, M,* and *N* fitted loosely when received and were fitted to a uniform $\frac{3}{32}$ in. crack and clearance before testing. Fig. 5 illustrates crack and clearance. It is important to note that the crack is one-half of the difference of the horizontal width between runways and the horizontal width of the sash. In the judgment of all who tried them, including a building carpenter boss and a construction inspector, these 5 windows were considerably looser than the average window in buildings. In the opinion of several weatherstripping mechanics, they were as loose as the usual loosely-fitted window they were called upon to weatherstrip in old build-

TABLE 2. FIELD MEASUREMENTS OF CRACK AND CLEARANCE IN INCHES

LOCALITY	NUMBER OF WINDOWS	AVERAGE FOR UPPER SASH		AVERAGE FOR LOWER SASH		MEETING RAIL	TYPE AND NUMBER OF BUILDINGS
		CRACK	CLEARANCE	CRACK	CLEARANCE		
MINNEAPOLIS, MINN.	91	0.0362	0.0465	0.0531	0.0520	0.0757	3 UNIV. - 2 RES. - 1 APT.
HARTLAND, WIS.	42	0.0395	0.0290	0.0603	0.0372	0.0471	5 RESIDENCES.
OCONOMOWOC, WIS.	16	0.0615	0.0415	0.0894	0.0264	0.0283	1 OFFICE
MILWAUKEE, WIS.	32	0.0220	0.0340	0.0720	0.0394	—	2 RESIDENCES
MILWAUKEE, WIS.	80	0.0984	0.0531	0.0765	0.0484	—	1 HIGH SCHOOL GROUP
MADISON, WIS.	16	0.0550	0.0527	0.0640	0.0733	—	1 UNIVERSITY
MADISON, WIS.	96	0.0373	0.0339	0.0659	0.0635	0.0804	6 UNIVERSITY
MADISON, WIS.	64	0.0373	0.0594	0.0694	0.0655	0.0847	1 UNIV. - 3 FRATERNITY
WATERFORD, WIS.	46	0.0615	0.0508	0.0544	0.0382	0.0577	3 RESIDENCES
MADISON, WIS.	64	0.0553	0.0510	0.0776	0.0905	0.0992	4 UNIVERSITY
MADISON, WIS.	32	0.0385	0.0437	0.0553	0.0609	0.0515	2 FRATERNITY
WEIGTED AVERAGE	TOTAL 579	0.0500	0.0455	0.0666	0.0567	0.0734	
EQUIVALENT IN FRACTIONS		3/64+	3/64-	1/16+	1/16-	5/64-	
CR & CL. USED ON 9 WINDOWS		1/16	3/64	1/16	3/64	—	

ings, although at times they encountered some that fitted more loosely. All sash would rattle badly except a few with sash members warped to such an extent as to prevent rattling. These 5 poor windows were tested as plain windows when the crack and clearance were first set at 3/32 in. and again 6 months later without any alteration to the fit except that which occurred in standing. They were then weatherstripped according to the schedule in Table 1 and again tested.

The 9 windows, *A, B, C, E, F, G, J, K,* and *L,* of a good grade were received fitted as closely as possible for free movement of the sash in opening and closing the window. They were tested after a standing period of 3 months. The crack was found to be approximately 1/64 in. and the clearance 1/32 in. By the definitions of crack and clearance, this means the windows had a tolerance of 1/32 in. in the two directions of sash fit; the difference of the distance between the runways and the width of the sash was 1/32 in. and between the width of the runway and the thickness of the sash was 1/32 in. In the judgment of those trying the windows, they were considered tighter than would be permissible in actual building construction, although under the conditions in the laboratory, they worked easily and smoothly. When they were retested after a standing period of 6 months from the date of the original tests, it was found that the crack and clearance had increased considerably. The measured crack and clearance were found to average slightly less than 1/16 in. and slightly more than 3/32 in. respectively. They were tested under this condition before a manual change in crack and clearance was made. During the entire 6 months, the windows were sheltered in the laboratory and were carefully handled in the few feet of moving to and from the machine and storage space. The original tests were made in a long dry period of the late summer, and the final tests were made under late winter and early spring heating conditions. After the original tests were completed and before the beginning of the heating season there was a rainy period when the humidity was considerably higher. The increase in crack and clearance during the 6 months' period was observed to be largely an opening up of joints in the frame, which increased the frame leakage and also allowed leakage that started at the outside of the sash perimeter to enter into the weight spaces. Some of this opening of frame joints was considered to be due to the warping of frame members and some to the effect on the frame members of the warping of the closely fitted sash members.

The 5 loosely fitted windows of a poor grade (*D, H, I, M,* and *N*) showed very little increase in crack and clearance and only a slight increase in leakage during the 6 months' standing period. This was considered to be due to the sash and frame members having sufficient room to warp without opening up joints and to the longer aging period of 1 yr instead of 3 months between the time of entering the laboratory to the time of testing. Before the final cracks and clearances were set for the final plain and the weatherstripped tests, the frame joints were closed by using larger nails with heads to supplement the finishing nails originally used.

Since in the judgment of those who tried the windows the 3/32 in. crack and clearance of the five windows of a poor grade were more than the average for old building construction, it was decided to actually measure the fit of a large number of windows. A total of 579 plain double hung wood windows was measured on buildings at least 5 yr old by about a dozen observers within

TABLE 3. RESULTS OF TESTS ON FIVE WINDOWS WITH 3/32 IN. CRACK AND CLEARANCE, PLAIN AND WEATHERSTRIPPED, IN CUBIC FEET PER HOUR, PER FOOT OF SASH PERIMETER

	WINDOW	CRACK	CLEARANCE	SASH PULLS POUNDS UPPER UP	UPPER DN.	LOWER UP	LOWER DN.	AVERAGE FOR 1ST AND 2ND CLOSURES — WIND VELOCITY–MILES PER HOUR 5.0	7.5	10.0	12.5	15.0	17.5	20.0	22.5	25.0	30.0	LOCKED — WIND VELOCITY–MILES PER HOUR 5.0	7.5	10.0	12.5	15.0	17.5	20.0	22.5	25.0	30.0
SEPT. TESTS	D	3/32	3/32	5	2	5	0	366	655	987	1311	1647	1986	2312	2620	2943	3577	134	300	517	766	1028	1318	1618	1923	2232	2710
	H	3/32	3/32	2	5	5	0	293	526	778	1028	1275	1522	1761	1992	2227	2682	194	337	489	649	794	928	1046	1160	1272	1489
	I	3/32	3/32	6	3	7	2	280	480	677	872	1067	1266	1465	1658	1861	2282	180	329	474	623	768	917	1065	1208	1360	1666
	M	3/32	3/32	5	2	4	2	230	433	646	868	1097	1315	1532	1754	1986	2468	111	231	374	537	712	894	1083	1280	1485	1904
	N	3/32	3/32	6	1	5	1	247	426	618	806	990	1176	1357	1047	1748	2182	100	209	349	503	665	846	1020	1189	1360	1700
	AVE. H & I	3/32	3/32	4	4	6	1	287	503	728	950	1171	1394	1613	1825	2044	2482	187	333	482	636	781	923	1056	1184	1316	1578
	AVE. M & N	3/32	3/32	5	1	4	1	239	430	632	837	1044	1245	1450	1650	1867	2325	106	220	362	520	689	870	1052	1235	1423	1802
	AVE. D,H,I,M,N	3/32	3/32	5	3	5	1	283	504	740	977	1215	1453	1686	1914	2153	2638	144	281	441	614	793	981	1166	1352	1542	1884
6 MO. LATER	D	3/32	7/64	5	2	6	0	440	743	1078	1372	1686	2037	2358	2683	3028	3752	280	523	803	1034	1294	1575	1872	2190	2543	3320
	H	3/32	3/32	4	3	6	0	356	620	882	1139	1394	1644	1904	2135	2440	3044	166	332	526	686	860	1037	1223	1412	1612	2050
	I	3/32	3/32	8	10	10	2	233	424	622	820	1020	1228	1443	1666	1894	2405	174	320	492	654	817	988	1168	1354	1557	2009
	M	3/32	3/32	5	2	4	2	214	429	649	871	1083	1297	1497	1700	1912	2355	157	326	508	703	892	1074	1242	1406	1571	1911
	N	3/32	3/32	4	4	5	1	303	494	689	869	1043	1212	1380	1560	1743	2147	163	332	526	720	917	1112	1298	1483	1669	2108
	AVE. H & I	3/32	3/32	6	6	8	1	295	522	752	980	1207	1436	1674	1899	2167	2725	170	326	509	670	839	1013	1196	1383	1585	2030
	AVE. M & N	3/32	3/32	4	3	4	1	259	462	669	870	1063	1255	1439	1630	1826	2252	160	329	517	712	905	1093	1270	1445	1620	2010
	AVE. D,H,I,M,N	3/32	3/32	5	4	6	1	309	542	785	1014	1245	1483	1718	1949	2203	2741	188	367	571	759	956	1157	1361	1569	1790	2228
WEATHERSTRIPPED	D			14	11	11	6	54	114	192	279	370	467	562	660	757	966	51	111	189	274	366	460	552	648	740	934
	H			12	9	14	6	57	127	180	250	306	377	452	536	631	834	57	127	180	250	306	377	452	536	631	834
	I			30	24	14	14	40	86	143	209	271	346	417	489	563	714	40	86	143	209	271	346	417	489	563	714
	M	AS	ABOVE	24	24	18	15	43	89	140	195	247	304	361	422	483	620	40	83	134	186	240	294	349	409	469	606
	N			19	20	15	8	40	86	140	200	259	322	389	461	536	695	37	83	134	192	252	312	374	440	514	660
	AVE. H & I			21	16	14	10	49	107	162	230	289	362	435	513	597	774	49	107	162	230	289	362	435	513	597	774
	AVE. M & N			22	22	17	11	42	87	140	197	253	313	375	441	510	655	39	83	134	189	246	303	362	425	492	633
	AVE. D,H,I,M,N			20	17	15	10	47	100	159	227	291	363	436	514	594	765	45	98	156	222	287	358	429	505	584	750

a 300-mile radius of Madison. On an average, 16 were taken on each building, 4 on each side. The readings were taken in December and January and therefore should represent cracks and clearances existing in the heating season. Eight of the men were assistants in the infiltration testing work and received personal instruction, and every observer received a complete and carefully worded description of the requirements and the method to be followed so that all values would be comparable. A blue-print defining crack and clearance, as does Fig. 5, accompanied the instructions as did mimeographed blanks to be used for the recording of data. Each observer was given a set of thickness gages for determining the clearance. The instructions emphasized the need for watching the warping of sash in measuring clearance and that the lack of clearance at the end of the meeting rails against the parting stops must be compensated for to obtain the full value of the crack. Particular emphasis was placed on the fact that the crack exists on both sides of the sash at the same time as shown in Fig. 5. The average values obtained were 1/16 in. crack and 3/64 in. clearance. The values obtained were slightly less than this for the upper sash and slightly greater for the lower sash. Table 2 shows a summary of the cracks and clearances obtained. The crack and clearance on the series of nine windows were accordingly adjusted to these values of 1/16 in. and 3/64 in. before the final plain and weatherstripped tests were made.

WEATHERSTRIPPING

After the crack and clearance had been set to the final values and the leakage of the plain windows determined, the windows were weatherstripped by local mechanics representing the weatherstrips they installed. The 3 mechanics for the 3 strips were men that had been at such work for many years and had had at least several years' experience with the strip they installed. Two interlocking and one rib type of strip were represented. The mechanics were instructed to install the strips just as they would in actual building construction and it is believed that they did this. The average time to install one strip was 1½ hr and the deviation from this was little. It was not necessary to increase the crack or clearance at the sides of the sash except in a few cases in order to square up the sash.

PRESENTATION OF RESULTS

The results of the tests on the 14 windows are shown in Tables 3 and 4 and in curve form in Figs. 6 to 13 inclusive. Table 1 explains the system of designating the windows and weatherstrips in the tables and on the curves. The tables in addition to giving the leakages at pressures corresponding to wind velocities up to 30 mph also give the pull in pounds necessary to open and close each sash in every test. These pulls give some indication of how tight a plain window has been fitted and the increase in sash pulls upon weatherstripping gives an indication of how closely a strip has been fitted. In some cases, a strip might be made very effective but at an undue loss of ease in opening and closing.

The tables present the values for individual windows and also averages for all of the windows of the two series termed as of good grade and poor grade to which each type of weatherstrip was applied and also the average of the entire groups of 5 and 9 of the two series. Although two not-locked runs,

TABLE 4. RESULTS OF TESTS ON NINE WINDOWS WITH 1/64 IN. CRACK AND 1/32 IN. CLEARANCE AND WITH 1/16 IN. CRACK AND 3/64 IN. CLEARANCE, PLAIN AND WEATHERSTRIPPED, IN CUBIC FEET PER HOUR, PER FOOT OF SASH PERIMETER

	WINDOW	CRACK	CLEARANCE	SASH PULLS POUNDS UPPER UP	UPPER DN	LOWER UP	LOWER DN	AVERAGE FOR 1ST AND 2ND CLOSURES WIND VELOCITY - MILES PER HOUR 5.0	7.5	10.0	12.5	15.0	17.5	20.0	22.5	25.0	30.0	LOCKED WIND VELOCITY - MILES PER HOUR 5.0	7.5	10.0	12.5	15.0	17.5	20.0	22.5	25.0	30.0
SEPT. TESTS – ORIG. FIT.	A	1/64	1/32	8	9	7	6	28	58	94	134	178	223	269	316	365	462	26	54	89	129	172	216	260	306	353	452
	B	1/32-	1/32+	7	8	2	3	20	46	80	119	160	236	257	311	369	494	22	52	89	132	179	230	285	343	405	535
	C	1/64	1/32+	15	11	7	8	21	45	77	113	154	198	244	286	336	433	19	43	74	110	151	195	240	286	332	428
	E	1/64+	1/32	5	3	3	4	33	68	108	155	204	258	315	376	440	569	33	68	108	155	204	258	314	373	435	562
	F	1/64	1/32	7	6	5	2	42	89	144	205	271	343	422	504	592	733	44	92	148	210	277	349	420	488	558	673
	G	1/64	1/32	16	20	4	6	16	36	62	92	127	166	205	248	293	387	10	26	46	71	97	129	163	200	238	320
	J	1/64-	1/32	24	24	11	10	20	45	77	115	155	199	244	291	340	440	17	39	68	103	141	182	225	269	315	411
	K	1/64-	1/32	5	7	8	8	42	84	131	183	238	294	351	409	470	601	42	84	131	183	238	294	351	409	470	601
	L	1/64	1/32	14	15	36	34	31	63	102	147	195	246	298	352	408	531	33	66	106	152	202	253	306	361	418	543
	AVE. A,B,C	1/64+	1/32+	10	9	5	6	23	50	84	122	164	219	257	304	357	463	22	50	84	123	167	214	262	312	363	472
	AVE. E,F,G	1/64+	1/32-	9	10	4	4	30	64	105	151	201	256	314	376	442	563	29	63	101	145	194	245	299	353	410	518
	AVE. J,K,L	1/64-	1/32	14	15	19	17	31	64	103	148	196	246	298	351	406	524	31	63	102	146	194	243	294	346	401	518
	AVE. C,G,L	1/64	1/32+	15	15	16	16	22	48	80	117	159	203	249	295	345	450	21	45	75	111	150	192	236	282	329	430
	AVE. A,B,C,E,F,G,J,K,L	1/64	1/32	11	11	9	9	28	58	96	140	187	240	290	344	402	517	27	59	96	136	185	234	285	337	391	502
6 Mo. LATER	C	1/32	3/64	22	20	12	10	46	98	154	214	274	340	420	497	580	763	34	76	129	180	246	306	386	463	540	723
	G	1/16	1/16	4	4	4	6	37	79	138	210	289	362	457	552	650	859	23	60	109	172	237	303	389	475	566	757
	L	5/64	3/64	7	7	16	16	50	113	185	260	330	409	495	578	666	856	51	117	189	266	337	414	500	583	672	866
	AVE. C,G,L	1/16-	3/64+	11	10	11	11	44	96	159	228	298	370	457	542	632	826	36	84	142	206	273	341	425	507	593	782
FINAL FIT	A	1/16	3/64	5	6	2	4	69	146	234	332	432	537	637	737	840	1043	80	163	251	354	454	560	663	766	866	1068
	B	1/16	3/64	4	5	2	4	50	103	168	235	311	382	476	563	656	872	50	106	172	237	315	389	483	572	666	874
	C	1/16	3/64	16	16	8	10	66	132	205	279	356	436	521	605	693	882	63	123	193	264	340	419	502	583	671	857
	E	1/16	3/64	4	2	3	3	51	119	196	285	380	473	572	671	773	996	56	123	203	300	406	515	623	731	849	1089
	F	1/16	3/64	6	4	6	5	74	149	240	337	440	534	652	757	866	1083	86	169	266	366	469	563	680	783	886	1091
	G	1/16	3/64	4	5	2	3	37	77	127	185	247	314	384	454	530	692	32	74	118	170	227	288	352	420	492	646
	J	1/16	3/64	10	12	8	10	62	125	205	291	386	486	585	690	796	1016	40	83	147	212	292	377	463	554	652	860
	K	1/16	3/64	4	5	3	6	46	92	163	240	321	395	489	577	663	845	43	89	160	234	315	386	477	560	640	812
	L	1/16	3/64	8	8	28	28	46	110	183	256	332	409	486	569	654	840	46	110	183	256	332	409	486	569	654	840
	AVE. A,B,C	1/16	3/64	8	9	4	6	62	127	202	282	366	452	545	635	730	929	64	131	205	285	370	456	549	640	734	932
	" E,F,G	1/16	3/64	5	4	4	4	54	115	188	269	356	440	536	627	723	924	58	122	196	279	367	455	552	645	742	942
	" J,K,L	1/16	3/64	7	8	13	15	51	109	184	262	346	431	520	612	704	900	43	94	163	234	313	391	475	561	649	837
	AVE. A,B,C,E,F,G,J,K,L	1/16	3/64	7	7	7	8	56	117	191	271	356	441	534	625	719	918	55	118	188	266	350	434	525	615	708	904

TABLE 4 (Concluded)

WEATHERSTRIPPED	A	1/16	3/64	33	32	10	10	2.2	3.9	6.4	9.1	12.0	14.9	18.0	21.3	25.0	32.7	1.6	3.4	5.5	7.9	10.2	12.7	15.5	18.3	21.4	28.1
	B	1/16	3/64	16	18	16	15	1.8	3.8	6.4	9.3	12.4	15.7	19.1	22.7	26.5	34.5	1.5	3.2	5.6	8.0	10.6	13.6	16.5	19.5	22.7	29.4
	C	1/16	3/64	22	22	13	12	2.1	4.4	7.2	10.3	13.8	17.2	20.7	24.1	27.6	35.5	1.7	3.5	5.7	8.4	11.3	14.2	17.2	20.2	23.5	30.4
	E	1/16	3/64	14	14	9	7	3.4	6.9	10.5	14.9	19.2	23.6	28.2	32.5	37.4	47.9	3.1	6.3	9.9	14.0	18.3	22.7	27.1	31.3	35.9	45.4
	F	1/16	3/64	18	16	12	9	4.1	8.3	13.0	17.7	22.5	27.4	32.6	37.9	43.8	56.8	3.6	7.6	12.1	17.2	21.7	26.0	30.7	35.4	40.6	51.6
	G	1/16	3/64	16	16	9	12	3.0	6.3	10.1	14.0	18.3	22.3	26.6	31.2	38.8	45.6	3.6	7.1	11.1	15.2	19.7	24.0	28.3	32.9	37.2	47.7
	J	1/16	3/64	22	24	16	15	2.8	5.7	8.7	12.7	16.6	19.9	24.2	28.3	32.4	40.7	2.6	5.4	8.3	12.3	16.0	19.2	23.4	27.4	31.4	39.7
	K	1/16	3/64	18	20	11	10	2.7	5.3	8.5	11.7	15.1	18.0	21.8	25.5	29.4	38.4	2.4	4.7	8.2	11.5	14.7	17.8	21.6	25.4	29.5	38.9
	L	1/16	3/64	17	15	22	20	1.9	4.0	6.7	10.3	13.9	17.9	22.0	26.0	30.5	35.2	1.7	3.4	6.0	9.1	12.6	16.3	20.0	23.7	28.3	37.4
	AVE. A,B,C	1/16	3/64	24	24	13	13	2.0	4.0	6.6	9.6	12.7	15.9	19.3	22.7	26.4	34.2	1.6	3.4	5.6	8.1	10.7	13.5	16.4	19.3	22.5	29.3
	" E,F,G	1/16	3/64	16	15	10	9	3.5	7.2	11.2	15.5	20.0	24.4	29.1	33.9	38.9	50.1	3.4	7.0	11.0	15.5	19.9	24.2	28.7	33.2	37.9	48.2
	" J,K,L	1/16	3/64	19	20	16	15	2.5	5.0	8.0	11.6	15.2	18.6	22.7	26.6	30.8	39.8	2.2	4.5	7.5	11.0	14.4	17.9	21.7	25.5	29.7	38.7
	AVE A,B,C,E,F, G,J,K,L	1/16	3/64	20	20	13	12	2.7	5.4	8.6	12.3	16.0	19.6	23.7	27.7	32.0	41.4	2.4	5.0	8.1	11.5	15.0	18.5	22.3	26.0	30.1	38.7

TABLE 5. FRAME LEAKAGE IN FRAME AND MASONRY WALLS IN CUBIC FEET PER HOUR PER FOOT OF SASH PERIMETER

PRESSURE DROP ACROSS WINDOW IN INCHES OF WATER	WIND VELOCITY MI/HR.	WOOD FRAME WALLS					MASONRY WALLS					
		NARROW PLASTER GROUND		WIDE PLASTER GROUND		AVE. OF FOUR TESTS	NOT CALKED			CALKED		
		WALL "A"	WALL "B"	WALL "A"	WALL "B"		WALL "A"	WALL "B"	AVE.	WALL "A"	WALL "B"	AVE.
0.012	5.0	3.2	2.6	2.5	2.3	2.7	6.2	1.9	4.1	0.8	0.4	0.6
0.027	7.5	5.9	5.0	4.5	4.5	5.0	10.2	3.8	7.0	1.5	0.9	1.2
0.048	10.0	9.0	8.0	6.6	7.0	7.7	14.3	6.3	10.3	2.3	1.5	1.9
0.075	12.5	12.3	11.0	8.8	9.7	10.5	18.4	9.2	13.8	2.9	2.2	2.6
0.108	15.0	16.0	14.4	11.0	12.7	13.5	22.7	12.3	17.5	3.6	2.9	3.3
0.147	17.5	20.3	18.2	13.7	16.0	17.1	27.0	15.6	21.3	4.3	3.7	4.0
0.192	20.0	24.9	22.2	16.3	19.5	20.7	31.5	19.1	25.3	5.0	4.4	4.7
0.243	22.5	29.7	26.3	19.1	23.1	24.6	36.3	22.7	29.5	5.7	5.0	5.4
0.300	25.0	34.9	30.8	22.2	27.0	28.7	41.6	26.3	34.0	6.4	5.6	6.0
0.431	30.0	46.4	40.5	28.9	35.5	37.8	53.1	33.5	43.3	8.0	6.4	7.2

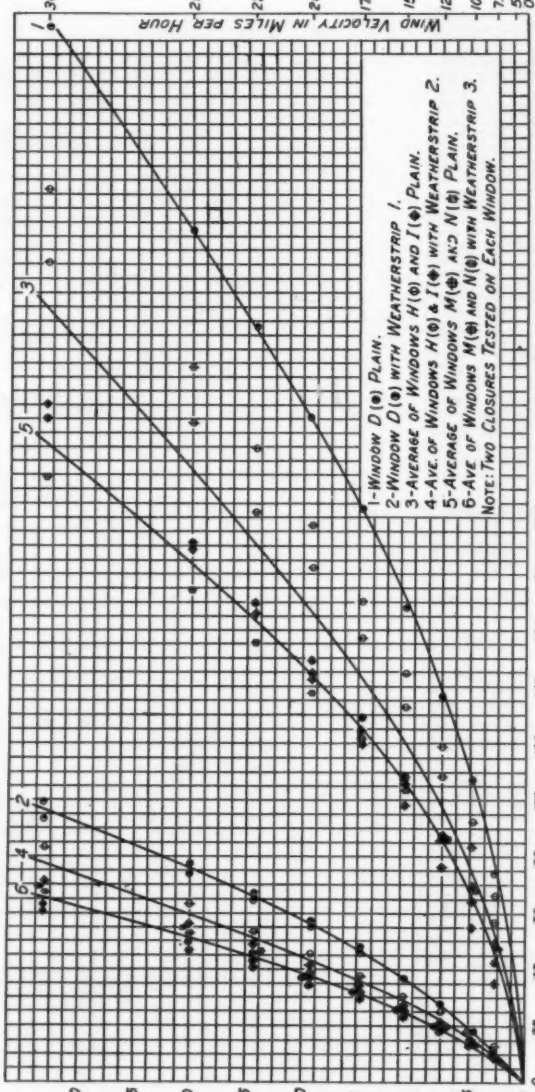

FIG. 6. RESULTS FOR 5 PLAIN AND WEATHERSTRIPPED WINDOWS WITH 3/32 IN. CRACK AND CLEARANCE—NOT LOCKED

termed first and second closures, were made in each case, the tables show only the average of the two. The curve sheets that show the points for individual windows and the averages of groups as curves show points for both the first and second closures except where they were identical. However, in getting values for average curves all test values, whether identical or not, were given equal weight. This explanation is considered necessary because some of the curves do not appear to be the average of the points they represent because some runs are represented by 1 point and others by 2 points at each pressure reading.

Table 3 shows the results of the tests on 5 windows of a poor grade with 3/32 in. crack and clearance, both plain and stripped. Table 4 shows the results for the 9 windows termed as of a good grade both plain with the original 1/64 in. crack and 1/32 in. clearance and plain and weatherstripped with the final 1/16 in. crack and 3/64 in. clearance that were established as representing the average crack and clearance by field measurements on a large number of windows.

Fig. 6 presents the results for the five windows plain and weatherstripped with 3/32 in. crack and clearance, not locked. Fig. 7 is the same for the locked runs. Figs. 8 and 9 show the corresponding results for the nine windows with 1/16 in. crack and 3/64 in. clearance. These 4 figures show the points by individual windows and the average curves by weatherstrips.

Fig. 10 shows the variation in infiltration over the standing period of 6 months between the original and the check tests without the making of a manual alteration in the crack and clearance. Fig. 11 shows the average curves for the series of 5 windows using 3/32 in. crack and clearance under locked and not-locked, plain and weatherstripped conditions. Fig. 12 shows the average curves for the series of nine windows with the original 1/64 in. crack and 1/32 in. clearance and with the final 1/16 in. crack and 3/64 in. clearance, both plain and weatherstripped.

DISCUSSION OF RESULTS

The average leakage found for the 5 plain, not-locked windows with 3/32 in. crack and clearance, was 124.5 cfh per foot of sash perimeter at 15 mph. This is the result of the tests made directly before weatherstripping and compares with 29.1 cfh for the same windows weatherstripped with the same crack and clearance. The points on Fig. 6 show that there was considerable variation in leakage between the 5 plain windows. This is considered to be due largely to the poor workmanship and material of this group. It was difficult to set an even crack and clearance on these 5 windows.

Window *D* showed a very high leakage as a plain window due to a combination of an upper sash that was warped and not square so that the head and meeting rail leakages were high and a larger than average clearance. Plain window *H* was also higher in leakage than the average when not locked. Locking reduced the leakage through this window so that windows *H*, *I*, *M*, and *N* had very nearly the same air leakage when locked as shown in Fig. 7. Locking window *D* reduced the leakage somewhat but left its position relative to the average about the same. Although window *D* deviates from the average considerably, it appeared to be an entirely possible window to be in any group of 5 windows of this type; consequently, it was used in the averaging of the group. The average leakage at 15 mph for this window when plain, for both

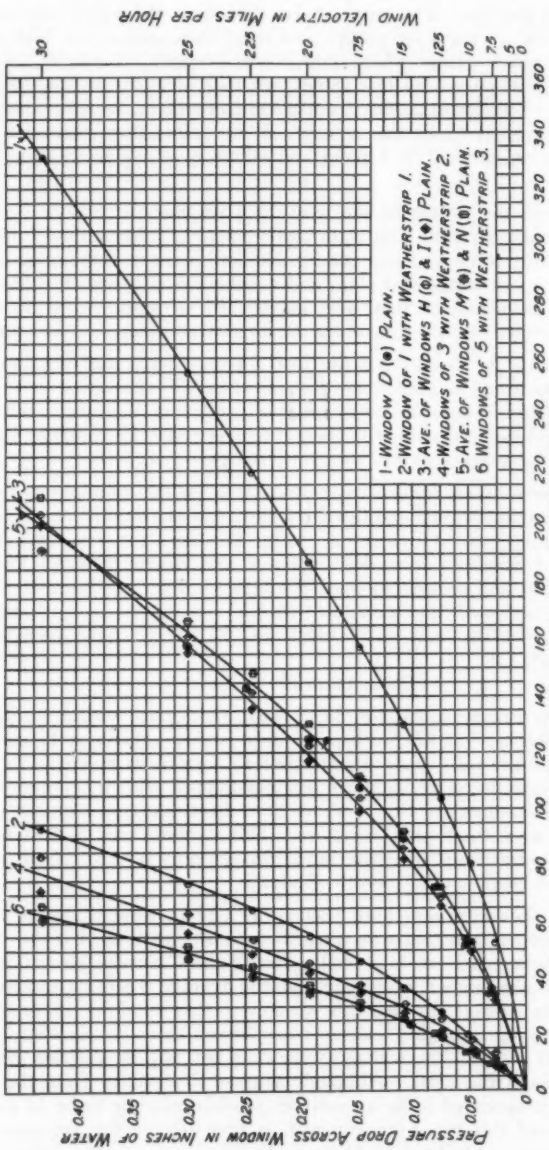
PRESSURE DROP ACROSS WINDOW IN INCHES OF WATER
0.40
0.35
0.30
0.25
0.20
0.15
0.10
0.05
0
INFILTRATION IN C.F.M. PER FT. OF SASH PERIMETER
0 20 40 60 80 100 120 140 160 180 200 220 240 260 280 300 320 340 360
WIND VELOCITY IN MILES PER HOUR
30 25 22.5 20 17.5 15 12.5 10 7.5 5 0
1-WINDOW D (●) PLAIN.
2-WINDOW OF 1 WITH WEATHERSTRIP 1.
3-AVE. OF WINDOWS H (⊙) & I (◆) PLAIN.
4-WINDOWS OF 3 WITH WEATHERSTRIP 2.
5-AVE. OF WINDOWS M (⊛) & N (▮) PLAIN.
6 WINDOWS OF 5 WITH WEATHERSTRIP 3.
1
2
3
4
5
6

FIG. 7. RESULTS FOR 5 PLAIN AND WEATHERSTRIPPED WINDOWS WITH 3/32 IN. CRACK AND CLEARANCE—LOCKED

locked and not locked runs, was 35 per cent above the average of the group. It was 28 per cent above the average of the group when weatherstripped. Windows *H* and *I* were 8 per cent lower than the average of the group as plain

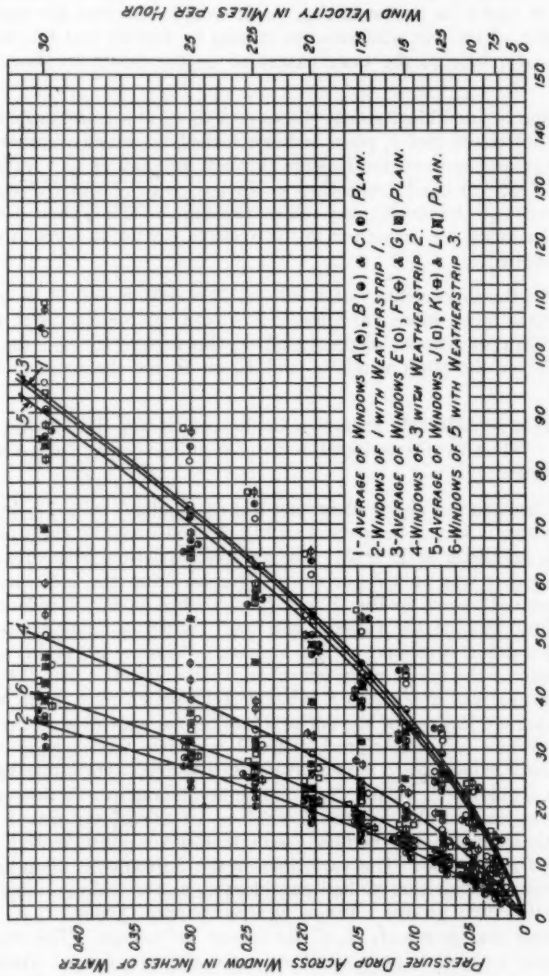

FIG. 8. RESULTS FOR 9 PLAIN AND WEATHERSTRIPPED WINDOWS WITH 1/16 IN. CRACK AND 3/64 IN. CLEARANCE—NOT LOCKED

windows and were equal to the average as weatherstripped windows. Windows *M* and *N* were 10 per cent lower than the average when plain and were 14 per cent lower when weatherstripped, considering both locked and not-locked results. This seems to indicate that the leakage through the weatherstripped

window for this class and fit of window depends largely on the leakage of the original plain window. The curves in Figs. 6 and 7 indicate this relation between the leakage of the plain and weatherstripped windows. Weatherstrip (1) applied to window *D* did not prove relatively as effective as when applied to windows *A*, *B*, and *C* as shown in Fig. 8. It would seem that the high leakage of the poorest of the five windows was caused by factors that the weatherstrip could not overcome to the same extent as on the better windows *A*, *B*, and *C*.

An examination of individual points shows that there is a considerable variation between windows *H* and *I*, plain and not locked, to which weatherstrip *2* was applied. Locking reduced the leakage through window *H* to a value very close to that of *I* when locked. Applying weatherstrips to *H* and *I* brought the leakage of these two windows very close together. The points for each weatherstripped window lie very close to the average for the windows to which that strip was applied.

The curves for the three groups of windows, *D* and *H I*, and *M N* in Figs. 6 and 7 show that much more variation exists between the averages for the plain windows than for the weatherstripped windows. At 15 mph the variation is 62 cu ft for the plain windows and 12 cu ft for the weatherstripped windows.

Fig. 11 shows a considerable reduction in the results for the locking of the plain windows. When weatherstripped, locking resulted in very little reduction on these 5 windows.

The average leakage for the 9 plain, not-locked windows with 1/16 in. crack and 3/64 in. clearance is shown in curve *3* of Fig. 12 and was 35.6 cu ft per hour per foot of sash perimeter at 15 mph. The greatest variation from this for the groups of 3 windows was only 3 per cent. The variation for individual windows in the group as shown in Fig. 8 was considerably more. Window *G* had an especially low leakage as a plain window. When weatherstripped, the leakage of this window was one of the highest. Window *A*, plain, was the highest in leakage of the entire nine, but when weatherstripped was the lowest. The same strip (1) was applied to a window (*C*) which as a plain window had about the average leakage of the entire 9 and to one with next to the lowest leakage (*B*). When weatherstripped, these 3 windows with widely varying leakages as plain windows were very uniform in leakage. The same variation is noticed with windows *E*, *F* and *G* to which weatherstrip (*2*) was applied. As plain windows, the results for the 3 windows of the group varied widely, yet when weatherstripped, the results group themselves quite closely to the average curve (*4* of Fig. 8). The variations in the 3 groups as plain windows are about the same except that *G* had considerably lower leakage than *B*, the lowest in the other two groups. When weatherstripped, group *E*, *F*, *G* was highest in leakage and group *A*, *B*, *C* the lowest in leakage. The results of the tests of these weatherstripped windows were closely grouped about their own average group curve (*2*, *6*, and *4*). This close grouping of the weatherstrips about the average curve for each group regardless of the variation of the individual windows in the group as plain windows, seems to indicate that each strip as applied to a window of this average crack and clearance has certain characteristics that determine the weatherstripped window leakage. Had

considerable variation been found among the weatherstripped windows of each group, it would have been due likely to a non-uniformity of the strip, to the influence of the plain window fit and leakage, or to the variation in workman-

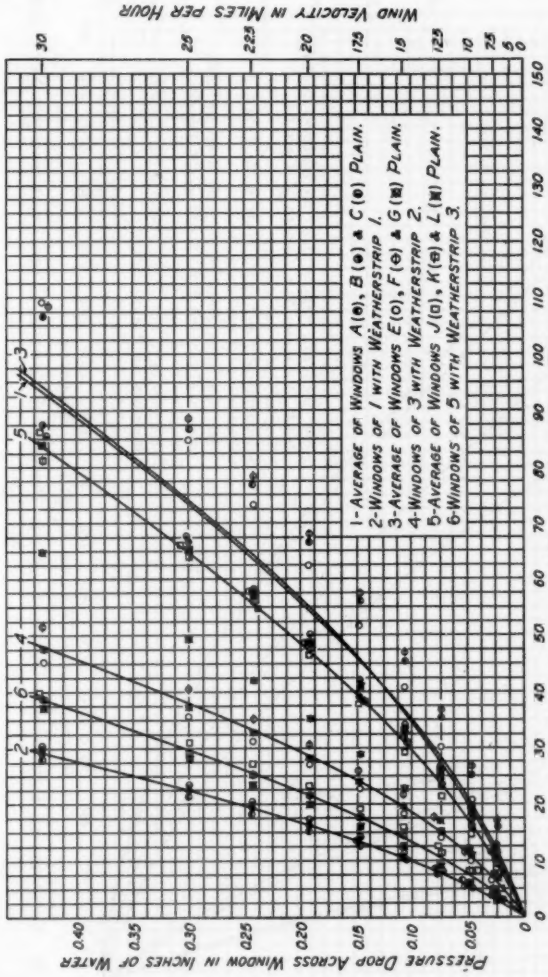

FIG. 9. RESULTS FOR 9 PLAIN AND WEATHERSTRIPPED WINDOWS WITH 1/16 IN. CRACK AND 3/64 IN. CLEARANCE—LOCKED

ship resulting when the same mechanic applied the same type of strip to several windows.

The grouping of the points for the individual weatherstripped windows *D*, *H*, *I*, *M*, and *N* with 3/32 in. crack and clearance as shown on Fig. 6 is very

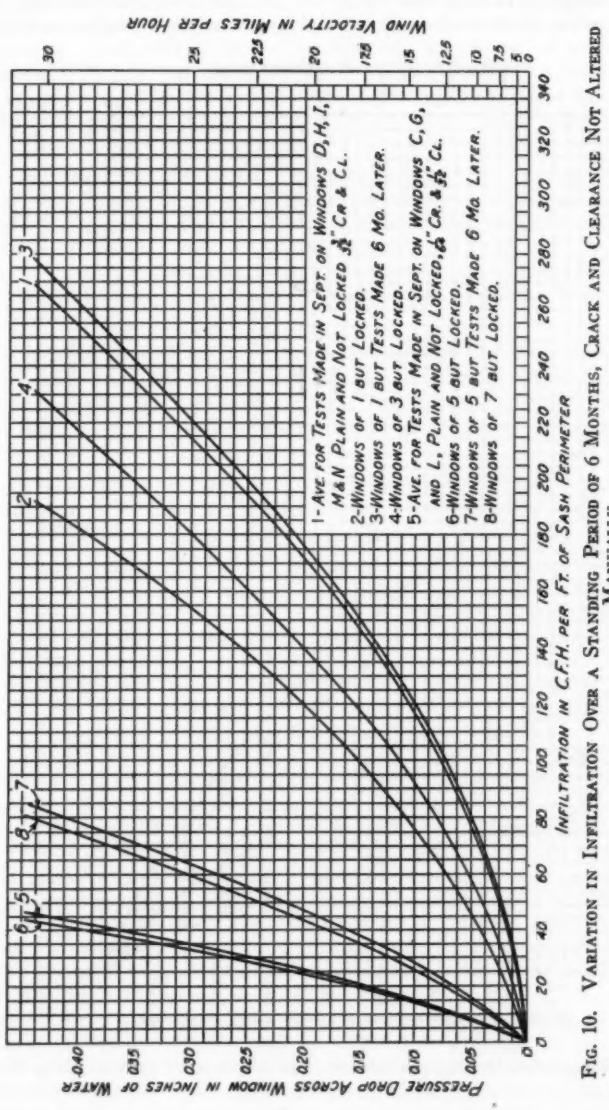
PRESSURE DROP ACROSS WINDOW IN INCHES OF WATER
0.40
0.35
0.30
0.25
0.20
0.15
0.10
0.05
0
0 20 40 60 80 100 120 140 160 180 200 220 240 260 280 300 320 340
INFILTRATION IN C.F.M. PER FT. OF SASH PERIMETER
WIND VELOCITY IN MILES PER HOUR
30 25 22.5 20 17.5 15 12.5 10 7.5 5 0
1-AVE. FOR TESTS MADE IN SEPT. ON WINDOWS D, H, I, M & N PLAIN AND NOT LOCKED. 1/32" CR & CL.
2-WINDOWS OF 1 BUT LOCKED.
3-WINDOWS OF 1 BUT TESTS MADE 6 MO. LATER.
4-WINDOWS OF 3 BUT LOCKED.
5-AVE. FOR TESTS MADE IN SEPT. ON WINDOWS C, G, AND L, PLAIN AND NOT LOCKED. 1/64" CR. & 1/32" CL.
6-WINDOWS OF 5 BUT LOCKED.
7-WINDOWS OF 5 BUT TESTS MADE 6 MO. LATER.
8-WINDOWS OF 7 BUT LOCKED.
1
2
3
4
5
6
7
8

FIG. 10. VARIATION IN INFILTRATION OVER A STANDING PERIOD OF 6 MONTHS, CRACK AND CLEARANCE NOT ALTERED MANUALLY

close to the average curves for *D* and *H I* and *M N*, although the plain windows *H* and *I* varied widely. One difference noted between the application of strips to the two groups of 5 and 9 windows was the relative positions occupied

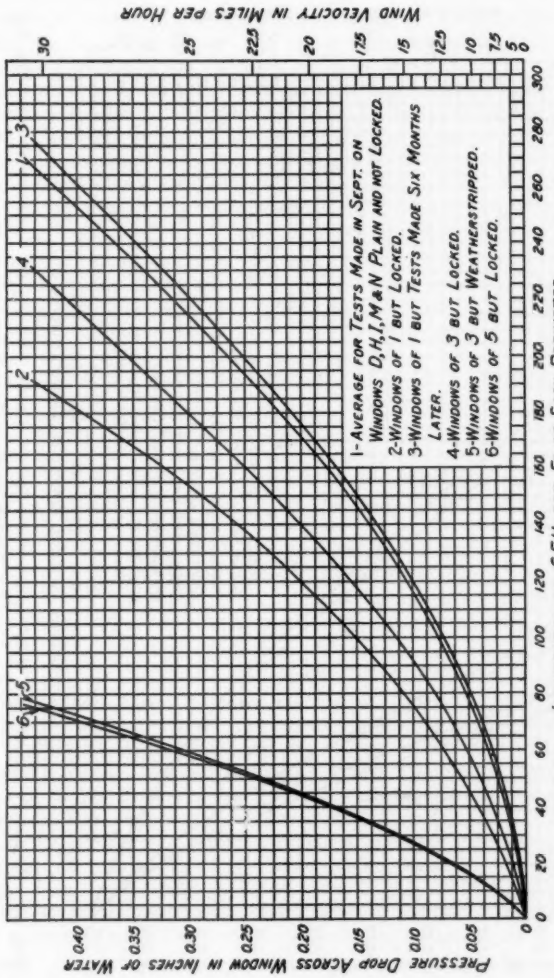

FIG. 11. SUMMARY CURVES FOR 5 WINDOWS WITH 1/32 IN. CRACK AND CLEARANCE

by the plain and stripped results of the windows *A*, *B*, *C* and *D* to which strip (1) was applied. *A*, *B*, and *C* as a group had the highest plain-leakage of the nine and as weatherstripped the lowest leakage. Window *D*, plain, had considerably the highest leakage of the 5 in the series, and when weatherstripped,

it still had the highest leakage. It, however, was about 8 per cent nearer the average of the group when weatherstripped than when plain.

The average leakage for the nine weatherstripped windows not locked as

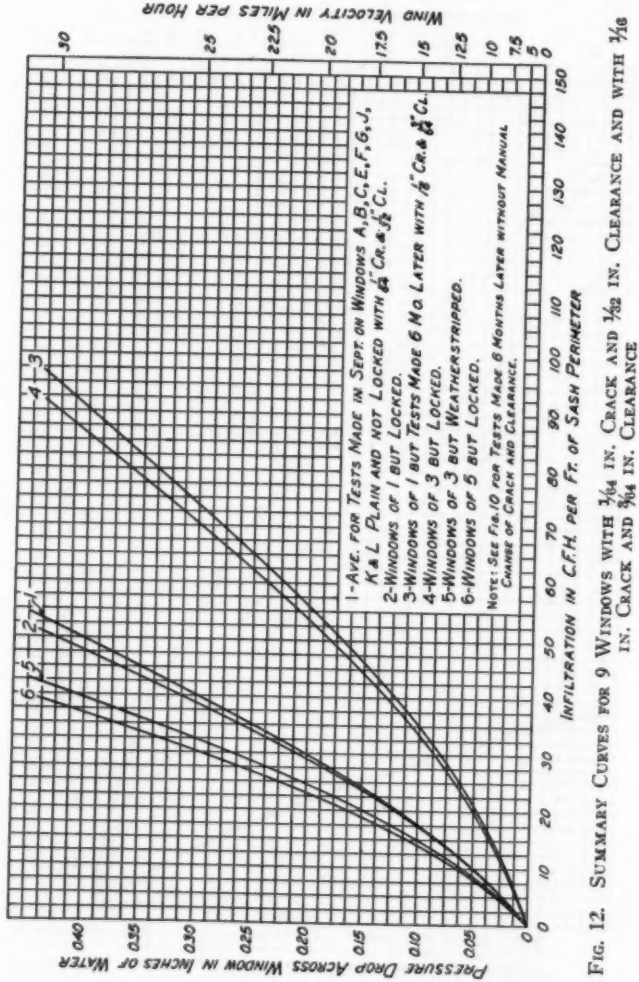

FIG. 12. SUMMARY CURVES FOR 9 WINDOWS WITH 1/64 IN. CRACK AND 1/32 IN. CLEARANCE AND WITH 1/16 IN. CRACK AND 1/64 IN. CLEARANCE

shown in Fig. 12, curve *5* was 16.0 cu ft per hour per foot of sash perimeter at 15 mph. The deviation of the average of each of the 3 groups from the general average of the 9 windows when weatherstripped is considerably more than the maximum deviation of 3 per cent when plain. Group *A, B, C* was

about 20 per cent lower than the average, group *E, F, G* was 25 per cent higher, and group *J, K, L* 5 per cent lower than the average. This indicates that there is more variation between the 3 groups when weatherstripped than when plain for these windows fitted to a 1/16 in. crack and 3/64 in. clearance.

The average leakage of the nine plain windows as originally fitted with 1/64 in. crack and 1/32 in. clearance was 18.7 cu ft as compared to 16.0 cu ft for the weatherstripped window with a crack of 1/16 in. and a clearance of 3/64 in. This is shown in Fig. 12. A plain window then can be made with a small crack and clearance and yet will open and close easily, and will have a leakage almost as low as a weatherstripped window fitted with an average crack and clearance. It is not known what the comparison would be if the weatherstrips had been applied to the windows retaining the small crack and clearance. It also must be borne in mind that the plain windows were fitted too closely to permit working under all conditions encountered in actual building constructions.

Locking the 9 plain and weatherstripped windows resulted only in a 1 cu ft reduction in leakage. The reduction due to locking was also slight in the case of the 5 windows when weatherstripped. The only case where locking resulted in a substantial reduction in leakage was that of these 5 plain windows which were fitted to a 3/32 in. crack and clearance. All locked runs were made with the locks just as they were originally fitted at the mill. Locking comparisons are shown on Figs. 11 and 12 for the two series of windows.

Fig. 10 shows the variation in results between the original tests made in September and 6 months later without manually changing the crack and clearance. On the 5 windows of a poor grade, the crack and clearance remained practically constant. The 6 months later tests on the plain, not-locked windows showed an increase in leakage of less than 3 per cent. Comparing the locked runs, the leakage of the same windows increased about 20 per cent. The reason for this larger increase for the locked as compared to the not-locked runs is not known. It would seem that a shrinking or warping of the window members would have less effect on the leakage when locked than when not locked.

Curve *5* of Fig. 10 shows the average leakage through 3 plain windows, *C, G,* and *L* with a 1/64 in. crack and 1/32 in. clearance. After a 6 months' standing period, the windows had become much looser. The average crack became slightly less than 1/16 in. and the average clearance slightly over 1/32 in. The average leakage for the plain, not-locked windows in the original tests was 15.9 cu ft per hour per foot of sash perimeter at 15 mph. The leakage after the standing period of 6 months was 29.8 cu ft. This is an increase in leakage of 87 per cent. A large part of the increase in clearance was due to the warping and shrinking of the wood pulling loose the finishing nails with which the frames were held together. The driving in of larger nails restored the clearances to somewhere near their original values. The crack and clearance were then adjusted to the final values of 1/16 in. crack and 3/64 in. clearance by planing. The leakage then obtained for the plain, not-locked window was 31.2 cu ft per hour per foot of sash perimeter at 15 mph. Locking resulted in a very small reduction in leakage in these 3 sets of tests on windows *C, G,* and *L.*

Locking resulted in no change in leakage in 20 per cent of all the runs made in these tests of the 14 windows. In another 20 per cent of the runs, locking

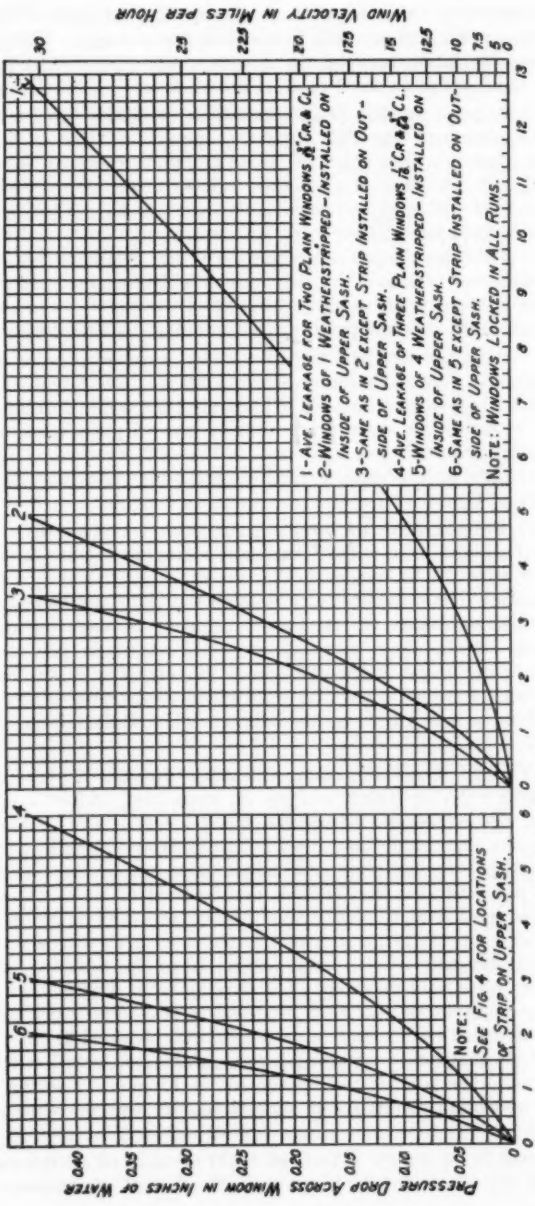
WIND VELOCITY IN MILES PER HOUR
PRESSURE DROP ACROSS WINDOW IN INCHES OF WATER
UNITS OF INFILTRATION
1-AVE. LEAKAGE FOR TWO PLAIN WINDOWS 1/8" CR. & CL.
2-WINDOWS OF 1 WEATHERSTRIPPED-INSTALLED ON INSIDE OF UPPER SASH.
3-SAME AS IN 2 EXCEPT STRIP INSTALLED ON OUTSIDE OF UPPER SASH.
4-AVE. LEAKAGE OF THREE PLAIN WINDOWS 1/8" CR & CL.
5-WINDOWS OF 4 WEATHERSTRIPPED-INSTALLED ON INSIDE OF UPPER SASH.
6-SAME AS IN 5 EXCEPT STRIP INSTALLED ON OUTSIDE OF UPPER SASH.
NOTE: WINDOWS LOCKED IN ALL RUNS.
NOTE: SEE FIG 4 FOR LOCATIONS OF STRIP ON UPPER SASH.

FIG. 13. EFFECT ON INFILTRATION OF STRIP LOCATION

increased the leakage and in 60 per cent of the runs, it decreased the leakage. The greatest decrease in leakage was 38 per cent and the greatest increase was 13 per cent. The average of all changes was an 8 per cent reduction.

Comparing the results of the first and second closures, it was found that in 31 per cent of the runs the leakage was the same for both closures. The average variation between the two closures was 5 per cent. The greatest variation was 21 per cent; the variation was greater than 10 per cent in 12 per cent of the runs.

Fig. 13 shows the influence on the infiltration of the location of the weatherstrip on the upper sash. Fig. 4 at *c* shows the strip located on the inside of the upper sash and *d* shows the strip located on the outside of the upper sash. When located as at *c*, it is possible for air entering the sash perimeter to gain entrance to the room through the pulley holes. The only reduction weatherstripping in this position could make in this leakage through the pulley holes would be in decreasing the clearance at the outside joint between the sash and the runway by exerting a force in that direction. The strip represented in Fig. 13 is regularly applied to the inside of the upper sash. The results of the tests of two windows with large crack and clearance weatherstripped in this manner are shown by curve *2* and as applied to 3 windows of average crack and clearance are shown in curve *5*. After the completion of these tests, the strips were removed from the upper sash and similar strips installed on the outside of this sash. The change in location resulted in a reduced leakage in every case. Curves *3* and *6* show the results for this relocation of the strips. The reduction was 26 per cent for the 2 windows with large crack and clearance and 37 per cent for the 3 windows with small crack and clearance. This reduction is due to the reduction of sash perimeter leakage that enters the room through the pulley holes.

FRAME LEAKAGE

The results of the tests on the 14 windows as given in Tables 3 and 4 and Figs. 6 to 13 inclusive are for sash perimeter leakage only. To this leakage must be added any air leakage that occurs through the frame of the window. In a plain plastered wall whether of masonry or frame construction, the leakage on a plain wall area the size of a window opening is negligible. Any leakage over this negligible amount when a window opening is placed in the wall is rightfully chargeable to the window. This leakage in a wood frame wall is mainly due to air entering at the edges of the building paper at the frame opening. A certain amount also enters at the corners of buildings and after getting under the paper travels in the joints of the sheathing from one studding space to another until a window opening, door opening or baseboard is reached that allows entrance to the room.

In a masonry wall, the leakage originates mainly at the joint between the window frame and the masonry wall, known as the frame joint. This leakage can be practically eliminated by the proper use of calking compound. A smaller amount of the frame leakage originates in the wall itself and travels horizontally in the voids to the window frame. Calking, of course, does not reduce this leakage. The use of solid masonry, that is, completely slushed joints, rather than the use of incompletely slushed joints that leave voids in the interior of the wall, would reduce this leakage. A portion of the frame leakage enters

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the room around the interior window trim, and some enters the sash weight boxes from where for the most part it enters the room through the lower sash pulley holes. The total pulley hole leakage is composed of this part of the frame leakage and the part of the sash perimeter leakage that entered the weight boxes mainly through the upper sash pulley holes. Pulley hole covers are at times applied to reduce this leakage. They reduce the very evident leakage at the pulley holes but the net reduction is not likely to be a large percentage.

FIG. 14. BOX FRAME IN MASONRY WALL TESTED FOR FRAME LEAKAGE

Their application likely builds up the pressure slightly in the weight spaces and the leakage for the most part finds its way into the room through smaller and less noticeable openings between the window trim and plaster or at the base-board. The proper place to stop the air leakage in good building construction would be at the points of entrance near the exterior surface of the wall.

Two 13-in. brick masonry wall sections and two wood frame wall sections were built to investigate frame leakages. Fig. 14 shows one of the masonry walls in the course of construction. Incompletely slushed walls were built in

each case but mortar was slushed completely against the window frames. The room sides of the walls were plastered. Fig. 15 shows one of the two frame wall sections completely built and in position ready to close the machine for testing. The construction from outside to inside was bevel siding, building paper, sheathing, 2 x 4 studding, wood lath and plaster. Two variations were tested on each of these two walls. In one, the plaster ground stopped at the 2 x 4 framing for the window opening; this was termed *narrow plaster ground* test. In the other, the plaster ground bridged across from the 2 x 4 window framing to the window frame so as to offer greater resistance to air passage

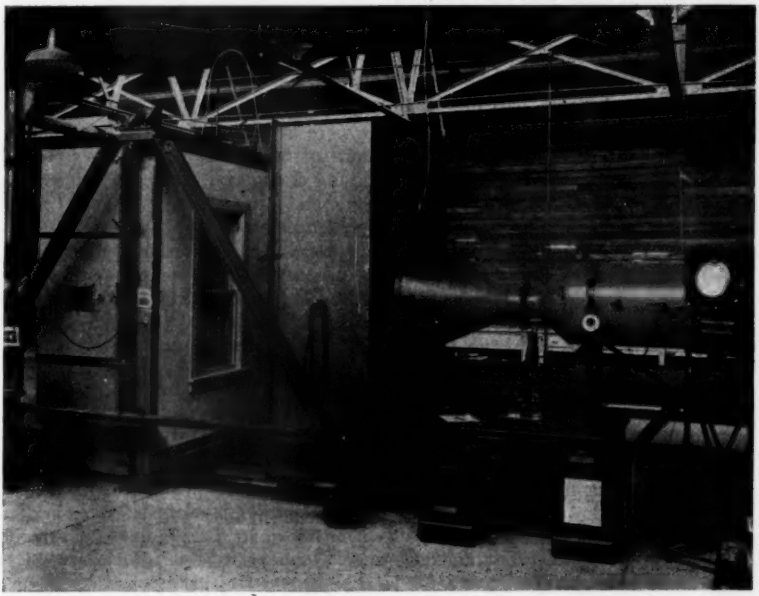

FIG. 15. WINDOW INSTALLED IN A WOOD FRAME WALL AND TESTED FOR FRAME LEAKAGE

from the studding and weight spaces and was termed *wide plaster ground* test. The wall sections were built by average building mechanics to represent average building construction.

Table 5 shows the results of these tests for frame leakage for both types of wall. The average leakage obtained at 15 mph for the wood frame walls was 13.5 cu ft per hour referred to per foot sash perimeter. The infiltration through the five plain, not-locked windows with $\frac{3}{32}$ in. crack and clearance was 124.5 cu ft and for these windows weatherstripped was 29.1 cu ft per hour per foot of sash perimeter at 15 mph. The average value for the 9 plain, not-locked windows with $\frac{1}{16}$ in. cracks and $\frac{3}{64}$ in. clearance which are considered to represent the fit of an average window was 35.6 cu ft and for the weather-

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stripped condition was 16.0 cu ft per hour per foot of sash perimeter at 15 mph. To these sash perimeter leakages should be added the 13.5 cu ft of frame leakage when applied to wood frame construction.

The average leakage through the frame in a masonry wall was found to be 17.5 cu ft per hour per foot of sash perimeter at 15 mph. Calking the frame joint resulted in reducing this leakage to 3.3 cu ft. When a perfect job of calking, which is easy to approach, is done, the frame leakage consists of air that enters into the brick work and travels to the window frame. This amounts to about 20 per cent of the uncalked frame leakage with the usual not completely slushed brick wall construction.

The particular frame leakage applying to the masonry wall, with the frame joint calked or not calked, should be added to the values of sash perimeter leakages to obtain the total leakage for the window opening.

Table 6 shows in the first line the average results for the plain and weatherstripped windows *D, H, I, M,* and *N* that were fitted to 3/32 in. crack and clearance. In addition, Table 6 shows the results of tests of seven windows termed *miscellaneous* of somewhat the same crack and clearance. These were tested as plain windows over the past several years and then weatherstripped. Only one test of each type is included in the miscellaneous tests although two different types of the same make are included in some cases. The thickness of the sash was 1 3/8 in. in every case. The results shown in this table are those from the original tests for the plain windows. The results shown as for weatherstripped windows are from tests made during 1931. The cracks and clearances are the measured values taken in the past at the time of the original plain window tests. Since the cracks and clearances are only approximately the same as those used on the present series of five, and because the method of sealing is in question on a few of the plain window tests and because of the cracks and clearances when tested as weatherstripped windows had probably changed from the original plain window values, it is considered best not to use an average obtainable from the inclusion of these so-called miscellaneous windows in Table 6. The average, allowing each window equal weight, however is given in this table for comparison to the average as determined in the present program on 5 windows of 3/32 in. crack and clearance. The average of the entire 12 is lower than the average of the 5 for both plain and weatherstripped conditions.

GENERAL CONCLUSIONS

Table 7 shows a summary of the frame leakages, the sash perimeter leakages and the sum of the two or the total window opening leakages for wood frame and for masonry wall constructions. In the case of masonry walls, only the total with the frame joint calked is given, since the benefit from calking is so obvious.

The Infiltration sections of A. S. H. V. E. GUIDES of 1930 and 1931 quote a leakage of 155.0 cu ft per hour per foot of sash perimeter at 15 mph. The crack is 1/16 in. and the clearance is 7/64 in. and the window not locked. The value for the weatherstripped window is given as 28.6 cu ft. A note states that these figures include the elsewhere leakage but not the frame leakage. These figures were arrived at by tests on windows with the seal made on the inside of the upper sash. The elsewhere leakage referred to as added is

TABLE 6. SUMMARY TABLE FOR INFILTRATION IN CUBIC FEET PER HOUR PER FOOT OF SASH PERIMETER THROUGH MISCELLANEOUS DOUBLE-HUNG WOOD WINDOWS WITH APPROXIMATELY 3/32 IN. CRACK AND CLEARANCE—NOT LOCKED

WINDOW	CRACK	CLEARANCE	PLAIN (6 MO. LATER)										WEATHERSTRIPPED									
			SASH PULLS POUNDS				MILES PER HOUR						SASH PULLS POUNDS				MILES PER HOUR					
			UPPER		LOWER								UPPER		LOWER							
			UP	DN.	UP	DN.	5	10	15	20	25	30	UP	DN.	UP	DN	5	10	15	20	25	30
AVE. FOR WINDOWS D,M,H,N,I	3/32	3/32	5	4	6	1	30.9	78.5	124.5	171.8	220.0	274.1	20	17	15	10	4.7	15.9	29.1	43.6	59.4	76.5
WINDOW O, WEATHERSTRIP 4	3/32	3/32	3	5	3	5	20.5	49.5	80.0	113.0	150.0	190.0	22	22	24	26	2.1	7.8	16.1	25.5	34.6	43.5
" P, " 5	1/8	3/32	7	4	4	2	27.5	74.8	126.0	175.5	223.0	266.0	18	16	10	4	2.4	10.0	20.5	32.5	45.0	57.0
" Q, " 6	1/8	3/32	-	-	-	-	14.5	50.0	81.0	116.0	147.0	175.0	8	6	10	7	5.1	18.0	34.4	52.0	68.6	85.3
" R, " 7	1/8	3/32	-	-	-	-	18.0	55.5	91.5	131.0	169.0	202.0	11	10	8	9	4.1	14.2	24.9	35.9	47.3	60.7
" S, " 8	7/64	3/32	2	1	9	2	25.0	58.0	91.5	122.5	152.9	181.0	15	11	25	20	5.0	14.8	25.2	36.0	48.2	62.0
" T, " 9	5/64	3/32	5	3	5	0	10.5	33.0	58.0	84.0	111.3	141.0	30	24	20	16	3.5	12.8	23.0	33.2	43.3	53.5
" U, " 10	5/64	3/32	5	5	8	0	8.7	27.3	49.3	72.5	97.3	123.5	34	34	40	28	2.6	9.7	19.7	31.2	42.8	54.0
WEIGHTED AVERAGE	3/32+	3/32	4	3	5	1	23.3	61.7	100.4	139.4	179.2	220.8	20	17	18	13	4.0	13.9	25.8	38.7	52.2	66.5

TABLE 7. SUMMARY TABLE OF INFILTRATION IN CUBIC FEET PER HOUR PER FOOT OF SASH PERIMETER THROUGH DOUBLE-HUNG WOOD WINDOWS—NOT LOCKED

PRESSURE DROP ACROSS WINDOW IN INCHES OF WATER	WIND VELOCITY ML/HR.	FRAME LEAKAGE			SASH PERIMETER LEAKAGE				FRAME LEAKAGE PLUS SASH PERIMETER LEAKAGE							
			MASONRY WALL		1/16 CR.& 3/64 CL.		3/32 CR.& 3/32 CL.		WITH WOOD FRAME WALL				WITH MASONRY WALL- CALKED			
									1/16 CR.& 3/64 CL.		3/32 CR.& 3/32 CL.		1/16 CR.& 3/64 CL.		3/32 CR.& 3/32 CL.	
		WOOD FRAME WALL	NOT CALKED	CALKED	PLAIN	WEATHER-STRIPPED	PLAIN	WEATHER-STRIPPED	PLAIN	WEATHER-STRIPPED	PLAIN	WEATHER-STRIPPED	PLAIN	WEATHER-STRIPPED	PLAIN	WEATHER-STRIPPED
0.012	5.0	2.7	4.1	0.6	5.6	2.7	30.9	4.7	8.3	5.4	33.6	7.4	6.2	3.3	31.5	5.3
0.027	7.5	5.0	7.0	1.2	11.7	5.4	54.2	10.0	16.7	10.9	59.2	15.0	12.9	6.6	55.4	11.2
0.048	10.0	7.7	10.3	1.9	19.1	8.6	78.5	15.9	26.8	19.3	86.2	23.6	21.0	10.5	80.4	17.8
0.075	12.5	10.5	13.8	2.6	27.1	12.2	101.4	22.7	37.6	22.7	111.9	33.2	29.7	14.8	104.0	25.3
0.108	15.0	13.5	17.5	3.3	35.6	16.0	124.5	29.1	49.1	29.5	136.0	42.8	38.9	19.3	127.8	32.4
0.147	17.5	17.1	21.3	4.0	44.1	19.6	146.5	36.3	61.2	36.7	165.4	53.4	48.1	23.6	152.3	40.3
0.192	20.0	20.7	25.3	4.7	53.4	23.7	171.6	43.6	74.1	44.4	192.3	64.3	58.1	28.4	176.3	48.3
0.243	22.5	24.6	29.5	5.4	62.5	27.7	194.9	51.4	87.1	52.3	219.5	76.0	67.9	33.1	200.3	56.8
0.300	25.0	28.7	34.0	6.0	71.9	32.0	220.3	59.4	100.6	60.7	249.0	88.1	77.9	38.0	226.3	65.4
0.431	30.0	37.8	43.3	7.2	91.8	41.4	274.1	76.5	129.6	79.2	311.9	114.3	99.0	48.6	281.3	83.7

leakage that originated at the sash perimeter and entered the room through the pulley holes. These figures, then, are to be compared to the sash perimeter results of the tests on the five windows with a 3/32-in. crack and clearance of the present program. The average sash perimeter leakage of these windows, plain, was 124.5 cu ft and 29.1 cu ft for the weatherstripped window per foot of sash perimeter per hour at 15 mph.

The measurement of a large number of windows in actual buildings at this time indicates that the crack and clearance of the average window is considerably less than the foregoing GUIDE values of 1/16 in. crack and 7/64 in. clearance or the 3/32-in. crack and clearance of the series of 5 windows of the present program. According to these measurements and the tests on the nine windows of this program, the average window would have a 1/16-in. crack and a 3/64-in. clearance and as a plain not-locked window would have a leakage of 35.6 cu ft per foot of sash perimeter per hour at 15 mph. When weatherstripped, its leakage on the average would be 16.0 cu ft. To these values should be added, when applied to wood frame construction, 13.5 cu ft of frame leakage, making a total of 29.5 cu ft per foot of sash perimeter per hour at 15 mph for the weatherstripped window. When applied to a calked masonry construction, the sash perimeter leakages should be increased by a 3.3 cu ft leakage making a total window opening leakage of 19.3 cu ft for the weatherstripped window. The plain window figures would be 49.1 cu ft for wood frame construction and 38.9 cu ft for masonry construction.

It is realized that the program reported in this paper is a laboratory program and that somewhat more representative figures might be obtained if a much larger number of windows were included in the tests. Furthermore, the effect of wear and weathering such as takes place in actual building constructions has not been considered. A valuable addition to the program as presented at this time might be made by the subjecting of the fourteen windows to wear and weathering. To accomplish this, they could be built into an enclosure that would be heated and on the outside subjected to weather conditions. The windows could be opened and closed a certain large number of times during a period of exposure of a year or more. The ease of opening and closing as measured by sash pulls under various weather conditions would check the belief of the authors that the weatherstrip installations on these fourteen windows represent average installation workmanship. The windows when removed from the exposure would be re-tested by the method followed in the tests of this paper to arrive at the leakage for a window as found in actual buildings.

DISCUSSION

W. C. RANDALL (WRITTEN): In view of the statement made in the next to last paragraph of the paper, do you feel that THE GUIDE should be revised to incorporate the infiltration secured from the good grade of window having 1/16 in. crack and 3/64 in. clearance as typical of general conditions? Since you were able to secure, from the mills, windows of such varying qualities, it is evident that all types are being used and it would seem rather risky to use a figure secured by testing good windows without knowing that this was the type being used in the particular building.

The tests from which data for THE GUIDE were secured showed very little difference in infiltration for crack widths from 1/16 in. up to 1/4 in. Since your tests found

the leakage for the good windows with 1/16 in. crack to be only about 1/3 of that for the poor ones with 3/32 in. crack (Table 7), would not this difference appear to be due to the type and construction of the window rather than the size of the crack?

Two major factors in weatherstripping are given in this paper. It has also been stated that windows are to admit light and to allow the entrance of air when it is wanted. Would not ease of operation, therefore, be a third important factor? It was somewhat disappointing to find, from Table 4, that the weatherstripping increased the pull required to open the windows to about five times that required originally. Could it not be said then that the most practical weatherstrip would be one that was efficient for preventing infiltration, without destroying the original ease of operation?

E. W. CONOVER (WRITTEN): Table 2, showing field measurements, lists the buildings mostly as university, high school and fraternity. Is it not thought that these represent buildings in which good windows had been installed rather than the average of buildings throughout the country? It appears doubtful that the five poor windows are typical of the average. Likewise, does it not seem reasonable that the nine good ones are above the average since it was stated that they were considered tighter than would be permissible in actual building construction?

The paper states that a plain window can be made that will have a leakage almost as low as a weatherstripped window fitted with an average crack for clearance. In view of the fact that this statement is based upon tests of the good windows and that these were considered tighter than permissible, also that six months later these same windows had a leakage about 80 per cent more than originally, is this considered a practical statement to make?

In regard to the frame leakage given in Table 5, I would like to ask whether or not the basis of per foot of sash perimeter is correctly stated. Ordinarily, sash perimeter includes the meeting rail which is not a factor in frame leakage. Should it not be per foot of frame perimeter?

When testing for frame leakage in different walls, it would be interesting to know how much time elapsed between building and testing, what grade of windows were installed, and what painting was done.

Is a continuation of this program, as outlined in the last paragraph, contemplated? If such is the case, would it not more nearly represent typical installation conditions if new windows are installed in a building and subjected to weather rather than the original 14 windows? New windows which have not been subjected to the seasoning that these original ones have might exhibit a greater tendency to shrink and warp as they weathered than would those that have been well seasoned in the laboratory.

AUTHORS' CLOSURE (WRITTEN): The original data for THE GUIDE were determined for crack widths of 1/16 in. to 1/4 in. The clearance was 7/64 in. Under such conditions, the crack makes very little difference as the clearance path is the narrower and therefore limits the flow of air. This clearance allows a 7/128 in. width of path on the sides of the sash as compared to a path 1/16 in. or more in width between the sash and the bottom of the runways.

In the present tests, the clearance was again the limiting factor rather than the crack. In the case of the average fit of windows the clearance path is 3/128 in. wide and in the case of the poor fit of windows, it was 3/64 in. Based on a direct comparison of these areas for air travel, it would be expected that the leakage would be double with the loosely fitted than with the average fit of windows. But doubling the width of the passageway more than doubles the effectiveness to air travel because of reduced skin friction due to a larger ratio of cross-sectional area to boundary surface.

Since this holds true, little of the difference in leakage of the so-called *good* and *poor* windows can be attributed to a difference in materials or workmanship. The

difference in the two types of windows was largely in the fit of the sash at the time of testing. The sash in each type were of the same material and grade. The frame material was poorer in the so-called poor windows in that poorer lumber was used. The parting stops also were of poorer material. But the two types of windows were probably made in the same types of mill work machines so that when adjusted to the same crack and clearance, the difference in material could affect the sash perimeter leakage very little. Unfortunately, no tests were made at the same crack and clearance. The windows were all ordered before the average crack and clearance were established by field measurements and the five poorly fitted windows were ordered loosely fitted as that was considered to go along with poor workmanship and materials. The nine closely-fitted windows were ordered to be as closely fitted as smooth working would allow because tests on a few such closely-fitted windows were desired and because naturally careful workmanship on the crack and clearance would demand the same grade of workmanship in other ways and the use of good material.

The paper has stated that there are two major factors in weatherstripping; one is the location of the strip and the other is the effectiveness of the strip to stop leakage that reaches it. Windows are to admit light and allow the entrance of air when it is wanted. This latter statement might be revised to say, to open easily to allow the entrance of air when it is wanted and to close easily when air is not wanted. It seems obvious that in improving the air resistance qualities of a window the weatherstrip should not affect adversely some other quality of the original plain leakage such as ease of opening.

Weatherstripping the nine windows of average fit resulted in an increase in sash pulls from about 7 lb to 16 lb, or a little over a doubling of the pull required to open and close. It seems natural that the installation of metal strips to definitely make close contact to reduce air leakage would increase the friction of sash movement. The only case where it would not increase the sash pull when properly fitted weatherstrips are applied would be with badly warped sash. Warping causes a drag on the plain window that is to a large extent corrected by the weatherstrip mechanic by truing with a plane at the time of strip installation. From this, it should not be understood that a moderate degree of warping is entirely a loss as it prevents rattling of sash in many plain windows.

The average sash pull of 16 lb when weatherstripped did not seem excessive and a period of operation would likely decrease this figure.

In obtaining the field measurements, an effort was made to secure windows in the average type of building using the double-hung wood type. Twelve of the 32 buildings were residences so that it seems doubtful that too much prominence has been given to university, high school and fraternity buildings. Further, some of the five fraternity buildings were at one time residences.

At no time were the five *poor* windows considered as typical of the average. They have been considered as representing the average of poorly fitted windows with a large crack and clearance. It probably is possible to find windows fitted more poorly; in fact, some may be so loose that the sash will not stay in the run-ways. But the five as fitted are in our judgment representative of the poorest fit that would be tolerated in a building to be heated to 70 F.

The nine windows termed *good* windows when fitted to the average crack and clearance as determined by field measurements were considered to represent the average window in a building at least five years old. When received as ordered, they were not average, since the crack and clearance were smaller than could be tolerated in a building exposure. These windows when tested plain with the final fit and when weatherstripped are taken to represent as closely as possible the average installed window. The only way of securing a closer figure to the average would be

by the testing of a large number of windows installed in buildings for a period of years.

The statement "that a plain window can be made that will have a leakage almost as low as a weatherstripped window fitted with an average crack and clearance" is a correct and practical statement to make when not divorced from the two succeeding statements, one of which states that "it should be borne in mind that the plain windows (when received) were fitted too closely to permit working under all conditions encountered in actual building constructions." The fact that the leakage increased by 80 per cent over a standing period of six months shows the original fit was not a practical fit. This increase in leakage was due to the letting go of the small finishing nails holding the frames together.

The results of tests for frame leakage as given in Tables 5 and 7 are correctly stated as per foot of sash perimeter. Sash perimeter has no direct relation or at least does not cause frame leakage, but the latter was tabulated on this basis for convenience in comparing with sash perimeter leakage and for direct addition in Table 7 to get total leakage per unit of sash perimeter chargeable against the window opening.

The brick walls were allowed to stand 2½ months and the frame walls one month between the time of building and testing. An average grade of window was used and the installations were made by average building mechanics to represent average building conditions. A prime coat was given all exposed wood on these walls.

The additional program suggested in the last paragraph of the paper of exposing windows to wear and weathering over a period of years was considered in connection with the 14 windows of the present program and now in the state of average weatherstripped windows. The additional exposure and retesting would prove our contention that these weatherstrip installations represent the average strip installation in buildings. Nine of these represent the installations on the average fit of windows and five on poorly fitted windows. It is the contention of some that a poor weatherstrip may be made *tight* on a laboratory installation but on exposure to weather the strips would not stand up nor would they retain their original freeness of movement. Exposure and wear tests would prove or disprove the author's belief that the strip installations are average workable installations.

Valuable additional data on double-hung wood windows might be secured by installing new plain windows in a building and subjecting them to wear and weather for a period of years before removing to a laboratory for test. This procedure was not contemplated in the writing of the last paragraph of the paper. One difficulty would be in setting the crack and clearance to such values that at the end of the period of exposure they will have assumed the values found for the average installed windows; that is, how much of the final crack and clearance as found on the average installed window is due to the original fit set by the building carpenter and how much to the wear and weathering of the following years. Another difficulty would be in the length of time needed to make exposure tests on the window plain and later weatherstripped. Likely the final leakage result is somewhat different, depending upon whether a weatherstrip is applied when the building is new or when it has aged for several years.

There is no end to the variations that might be studied. The present paper is considered to have determined a reasonably close value for the leakage through the average plain and weatherstripped window. The clearance of the average window was found to be $\frac{3}{64}$ in. and the crack $\frac{1}{16}$ in. It is considered that the figures obtained for these windows should be used in THE GUIDE to represent the average window of this type. In addition, the leakage was determined through poorly fitted windows having a $\frac{3}{32}$ in. crack and clearance. The results of these tests might also

well be incorporated in THE GUIDE. It would be up to the designer to establish whether the windows he is concerned with are average in fit or typical of poorly fitted windows. An average fit of windows is easy to obtain by specification and inspection just as is an average good brick wall. There would seem to be no reason for saddling an average window installation with excess radiation just because it is possible to make and install windows of poorer than average fit.

No. 910

PREDETERMINING THE AIRATION OF INDUSTRIAL BUILDINGS

By W. C RANDALL[1] AND E. W. CONOVER,[2] DETROIT, MICH.
MEMBERS

THE method presented in this paper for predetermining airation (the flow of air into, through and out of buildings due to the natural forces, the wind and temperature difference) is the result of efforts to shorten the somewhat lengthy but more technically correct method developed in the research investigations conducted under the direction of Prof. J. E. Emswiler of the Department of Engineering Research of the University of Michigan.

The correct method is to combine the heads created by these two forces at each window opening, or group of openings, occurring in the different zones of pressure, suction, and normal atmospheric pressure. These various zones are determined by the length, width and height of the structure and the direction of the wind, and may also be influenced by the distance from and size of neighboring buildings. Such factors must and still will be used when greater accuracy than can be secured by the simplified method is essential or desired. That a shorter method could be used was clearly indicated after solving several hundred problems. The method suggested herein was found to be reasonably accurate, at least within the percentage of error of other assumptions that have to be made because of lack of knowledge, or because of efforts to simplify the problem.

It is hoped that the method presented in this paper will make it possible to estimate airation in advance of construction with sufficient accuracy for practical purposes. It should be possible thereby to design the fenestration for any desired movement of air, and to make comparisons of the probable results from more than one design.

It is not to be presumed that this can be done accurately for every situation that arises. However, it can be done for certain simpler cases, and the performance under more complicated circumstances inferred with better judgment.

The forces of nature are always present, whether it is desirous that they be put to work or not. A better knowledge of the natural flow of air may explain

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some of the idiosyncrasies of mechanical systems, the success or failure of which depend to some extent upon whether or not the natural forces are co-operating with or opposing them, and as a result, more dependable installations can be designed.

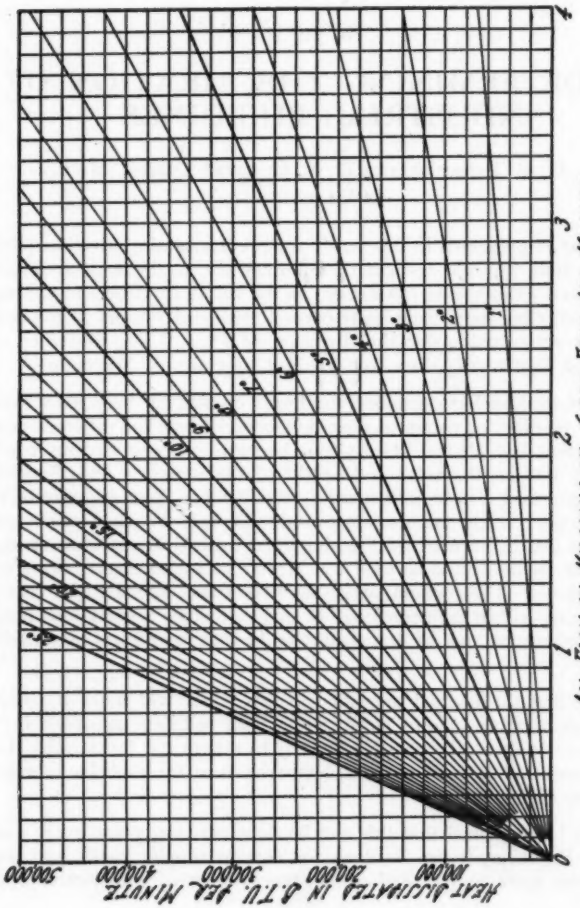

FIG. 1. AIR FLOW REQUIRED TO MAINTAIN DESIRED TEMPERATURE DIFFERENCE

In outline, this simplified method consists of three steps: (1) a determination of the air flow due to temperature difference, (2) the flow due to the wind, and (3) a combination of the flow created by these two forces acting together.

FLOW DUE TO TEMPERATURE DIFFERENCE

The air within a building absorbs much of the heat used in or generated by manufacturing processes, and must be carried off and be replaced by cooler outside air if conditions are to be maintained which will allow the workmen to continue their labors without undue discomfort. This is usually accomplished by providing upper openings through which the heated air may escape and lower openings through which the cooler air may enter.

The factors which influence the volume of air moved by temperature difference alone are:

1. The amount of heat given off.
2. The vertical distance between inlet and outlet.
3. The areas of inlet and outlet openings.

These three factors were adequately discussed in a previous paper[8] based upon this research work and a method was given therein for determining the air flow through a building induced by this force alone.

The difference in temperature existing between the inside and outside air is determined by the amount of heat given off inside the building, and the amount of air flowing through it. The first requirement, then, is to estimate the amount of heat to be carried off by the air in, for example, Btu per minute. This can be determined, in the case of forge shops and the like, from the amount of fuel burned which, in turn, is based upon the production capacity for which the building is being designed. In the case of foundries, the heat given off by the metal in cooling from the molten state can be used. In some cases, not all of the heat may be dissipated to the air, such as that carried off through stacks, that absorbed by the sand in foundries and that taken out by removing castings from the building while still hot. But a fair estimate of the amount to be removed by the air can usually be made.

The next step is to select the temperature difference to be maintained. The selected inside temperature should be low enough for the comfort of the workmen. In the summer time, when outside temperatures are high, the inside-outside temperature difference must not be great. Consequently, the greatest amount of air must be passed through the building at that time, and the airation characteristics determined for that condition.

Knowing the amount of heat to be removed and having selected a desirable temperature difference, the amount of air to be passed through the building per minute to maintain comfortable conditions can be determined by means of the following equation:

$$H = 0.0175QD \tag{1}$$

where

H = heat in Btu per minute
Q = air flow in cubic feet per minute
D = inside-outside temperature difference, degrees Fahrenheit

Equation (1) is solved graphically in Fig. 1, thus making it possible to determine readily the air flow necessary to maintain any desired temperature difference for a given heat emission.

[8] The Neutral Zone in Ventilation, by J. E. Emswiler (A. S. H. V. E. TRANSACTIONS, Vol. 32, 1926).

For a given temperature difference (D) and vertical distance between openings, the outlets will exhaust a definite amount of air. The elemental building dimensions, such as width, length and height, are usually determined by the nature and the amount of the product and the methods of handling production. Therefore, the approximate location of the windows is fairly well established, and the height of the outlets above the inlets is known.

It was stated in a previous paper,[4] that the inlets and outlets should be about equal for the greatest flow per unit of combined areas and, since this is the

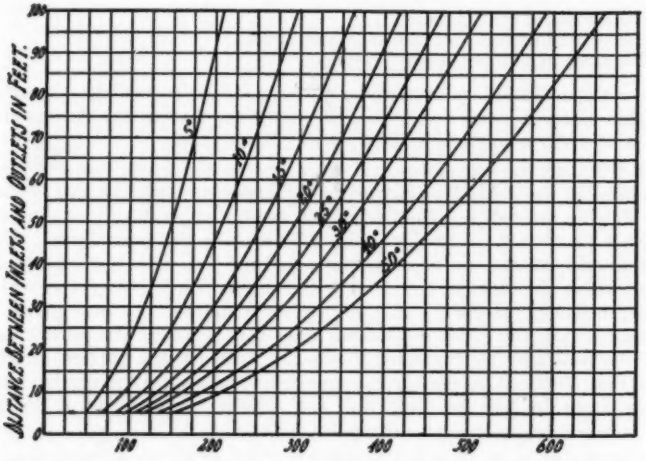

FIG. 2. FLOW PER SQUARE FOOT OF INLET OR OUTLET IN CUBIC FEET PER MINUTE—INLET AND OUTLET AREAS EQUAL

ideal construction, this condition will be considered first. Knowing the temperature difference to be maintained and the vertical distance between inlets and outlets, the flow per square foot of inlet or outlet can be determined by the following equation:

$$Q = 60\sqrt{\frac{MD}{2RT}} \qquad (2)$$

where

Q = flow in cubic feet per minute per square foot of inlet or outlet (when of equal area)

M = vertical distance between inlets and outlets in feet

D = inside-outside temperature difference, degrees Fahrenheit

T = average absolute temperature (460 F plus average between inside and outside, say 550 F for summer)

[4] Airation of Industrial Buildings, by W. C. Randall (A. S. H. V. E. TRANSACTIONS, Vol. 34, 1928).

R = resistance of opening $= \dfrac{1}{2gA^2C^2}$

g = acceleration due to gravity = approximately 32.2 ft. per second per second

A = opening area in square feet (in this case 1)

C = coefficient of resistance of the opening (0.65)

Substituting the foregoing values of R and T in equation (2),

$$Q = 9.4\sqrt{MD} \qquad (2a)$$

Fig. 2 has been plotted from equation (2a), and from this figure the flow per square foot of inlet or outlet opening, when of equal areas, may be read directly

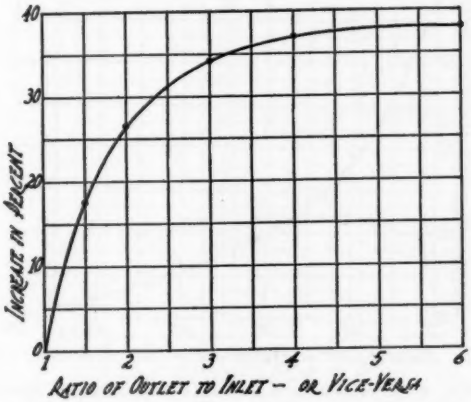

FIG. 3. INCREASE IN FLOW CAUSED BY EXCESS OF ONE OPENING OVER THE OTHER

for any given distance between the openings and any temperature difference. If for a certain building (1) the amount of air to be passed through, (2) the temperature difference to be maintained, and (3) the distance between inlets and outlets, are known, the amount of window opening to be installed as inlets (with a corresponding amount for outlets) may be ascertained by dividing the flow per square foot of opening, as determined from Fig. 2, into the total amount of air necessary to be moved.

Although the greatest flow per unit area of opening is obtained when inlets and outlets are equal, it is frequently found that a building has greater possible outlet openings than inlets, or vice versa. Equal openings will result in a certain amount of air flow. Increasing the area of one over the other will increase this flow, but not in proportion to the added area. The increase in flow produced by excess of one opening over the other is indicated by Fig. 3. For example, if the outlet area is double that of the inlet, the flow will be about 26½ per cent greater than would be the case if it equaled the inlet. This

curve has been obtained by solving problems according to the method outlined in the paper The Neutral Zone in Ventilation previously mentioned, thus giving a substitute for determining the neutral zone when openings are unequally divided.

When designing a building, it might be found that the amount of inlet opening calculated to be necessary could not be incorporated in the plans without major changes. Sometimes it is possible to lower the level of the inlets or raise that of the outlets, or both, which would give an increased head and a consequent increased flow per square foot of opening. If this cannot be done, then the alternative is to install as many lower sidewall openings as possible

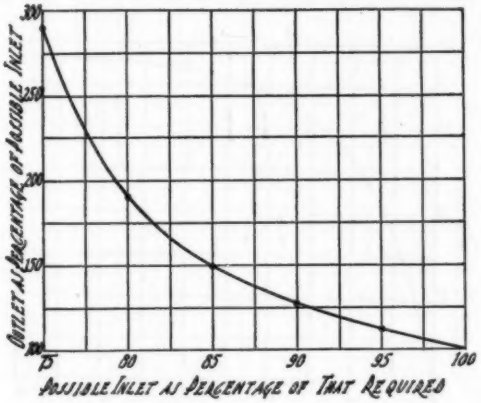

FIG. 4. COMBINATIONS OF INLET AND OUTLET OPENINGS TO PRODUCE THE SAME RESULTS AS FOR EQUAL INLET AND OUTLET OPENINGS

for inflow, and to increase the upper outlet openings to give the required flow, or vice versa.

Fig. 4 shows the combination of inlet and outlet areas that will give the same results as those secured by equal openings. For example, assume that 1,000 sq ft of inlet openings and an equal amount of outlets are necessary to give the desired results in a certain instance. On laying out the plans, it is found that but 800 sq ft of opening can be accommodated in the lower sidewall. This being 80 per cent of that required, it is found from Fig. 4 that the upper openings will need to be 190 per cent of 800 or 1,520 sq ft. This gives a total of 2,320 sq ft required for the equivalent performance of 2,000 sq ft equally divided between inlet and outlet.

FLOW DUE TO THE WIND

In machine shops, assembly buildings and the like, where little heat is generated, the airation is produced chiefly by the wind. The requirements for such

buildings are somewhat less than for foundries, forge shops, heat-treatment buildings, and similar types of buildings where smoke, gases, etc., prevail. When the wind is producing the airation, many factors enter into the correct solution of the problem. For simplification, the buildings would first be divided into two classes, namely, those with a monitor, or monitors, and those without, since there is an increased flow caused by monitors which usually place openings in the suction zones created by the wind hopping over the building.

The main factors which influence the air flow are the size and shape of the building, the velocity and relative direction of the wind and the size and location of window openings. It is believed that a simple and workable method must be based upon the average results produced by these factors.

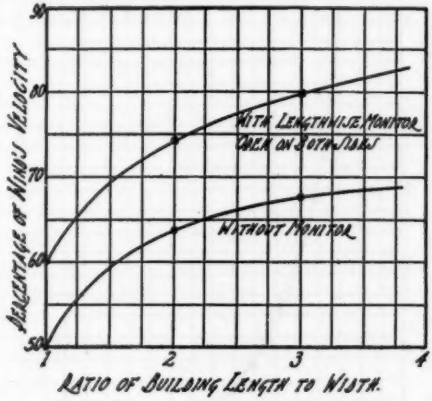

FIG. 5. PERCENTAGES FOR WIND AGAINST END

To determine the window openings required in a given building, it is necessary to select first the desired air changes or air movement. It is also necessary to know the velocity and direction of the prevailing winds for the particular location being considered. These usually can be obtained from the nearest local office of the U. S. Weather Bureau.

With this information available, the amount of window opening to be provided can be determined by simple percentages. If the building has no monitor and the wind will blow perpendicular to the side, the area of the openings to be installed in the windward side may be found by dividing the desired air flow in cubic feet per minute by 50 per cent of the velocity of the wind in feet per minute. If blowing diagonally, 30 per cent should be used, but this will give the combined area in one side and one end. Naturally, corresponding amounts will be necessary in the opposite side and end to act as outlets.

If a monitor is contemplated, this will put additional openings in the suction zone caused by the wind jumping over the building and, consequently, the air flow will be increased. If the monitor openings correspond in area to those

in the sidewall below, then the desired flow should be divided by 60 and 35 per cent of the velocity of the wind, respectively, for wind directions perpendicular and diagonal to the side. If a greater area is provided in the monitor than in the sidewall, these percentages will be increased and, if less, they will be decreased.

The foregoing percentages usually will take care of all cases when considering side or diagonal winds against all buildings not obstructed by others. If the prevailing wind will blow against the end of the contemplated building and

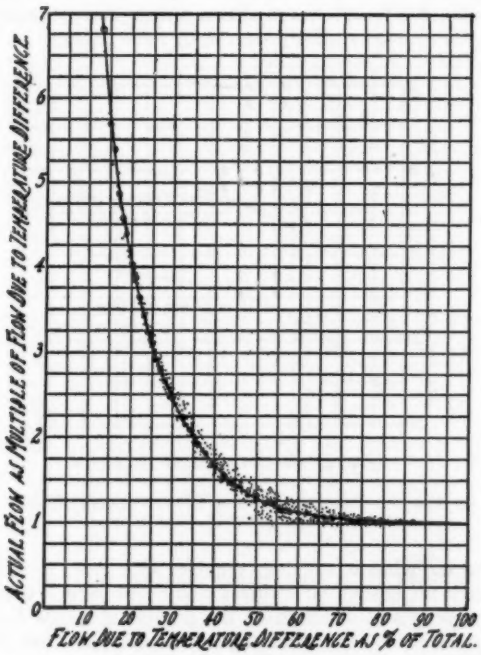

FIG. 6. DETERMINATION OF FLOW CAUSED BY COMBINED FORCES OF WIND AND TEMPERATURE DIFFERENCE

the structure is other than a square one, the percentages given by Fig. 5 will apply.

When end winds are blowing, it usually is found that some sidewall openings, other than those in the direct path of the wind, will provide inflow. This is due to a greater demand for air at openings in the areas of suction than can be supplied by the windward openings, thus pulling air in at openings beyond the direct influence of the wind. This will occur at monitor openings as well as sidewall, but in arriving at the prediction figures shown, a deduction has been made for what is considered ineffective inflow at monitor openings on the

assumption that some of it is short circuited and of no material benefit in the working zone. Thus, in predetermining the air flow by this method, the results represent more closely the effective flow rather than the total air passing through the building.

The curves of Fig. 5 are based upon the ratio of the length and width of the building, and will give the area of the windows to install in the end elevation. A corresponding fenestration in the bays of the sidewalls and monitor, if any, will complete the design.

FLOW DUE TO COMBINED WIND AND TEMPERATURE DIFFERENCE

To determine the size of window openings that will provide a predetermined flow of air through a building in which considerable heat is produced, it is desirable to use the method described for flow due to temperature difference alone, in order that adequate openings will be provided to take care of the days when little or no wind is blowing.

In some instances, this will show an amount that is impossible or impractical to install. Account must then be taken of the action of the wind and the design based on the combined forces. It is recognized, as previously stated, that the correct method is to combine the forces or heads caused by each, rather than to add the flows. However, it was found that there was some relation between the ratio of the calculated flows produced by each agent separately and the flow computed by adding the heads.

Fig. 6 shows that the flow created by the combined forces is a multiple of that due to temperature difference alone. This graph represents the results obtained chiefly from solving problems involving three building heights, five wind velocities, four temperature differences, three building lengths and all wind directions. It has been found that this method will give results that agree, in most cases, with those obtained by combining the forces, the maximum variation being about 15 per cent.

To apply this method, it is necessary first to compute the flow that would be produced by each force if acting alone. The percentage each is of the sum is then determined. Using the temperature-flow percentage as an index, the flow due to the combined forces which is a multiple of that calculated for temperature difference alone, may be obtained from Fig. 6. For example, assume that the computed flow due to temperature difference alone is 30 per cent of the added amounts and that due to the wind, 70 per cent. The flow due to the combined forces, as found from Fig. 6, would be 2.45 times that computed for temperature difference alone.

With the greater flow of air caused by the two forces acting together, it is evident that more heat will be carried out of the building and, consequently, the inside temperature will drop below that experienced when no wind is blowing. The magnitude of this drop can be determined from Fig. 1. For example, assume that the heat to be dissipated by the air is 200,000 Btu per minute, and that the windows are designed to maintain a temperature difference of 15 deg. The air flow induced by temperature difference alone is 750,000 cfm. Assume also that the effect of the wind increases this to 1,000,000 cfm. The amount of heat dissipated is not changed so the intersection of the "200,000 Btu" line with the "1,000,000 cfm" line gives the resultant inside temperature

as about 11½ deg above that outside instead of the 15 deg when no wind is blowing.

SOLUTION OF A TYPICAL PROBLEM

A typical problem will illustrate this procedure. Let it be assumed that the building outline has been determined to be as shown in Fig. 7, and that the prevailing conditions are as follows:

Drop forge shop
Building dimensions: 100 ft × 200 ft × 30 ft
Cubical contents: 600,000 cu ft
Height outlet above inlet: 30 ft
Fuel: 30 gal fuel oil per hour
Desired temperature difference: 10 deg
Wind diagonal at 8 mph

Computations for Temperature Difference Only

$$\text{The heat to be removed} = \frac{30 \text{ gal} \times 7.75 \text{ lb} \times 18{,}000 \text{ Btu}}{60} = 69{,}750 \text{ Btu per minute.}$$

This amount of heat will require about 400,000 cfm of air (Fig. 1) or 40 air changes per hour to maintain the desired temperature difference. With 30 ft between inlet and outlet and a 10-deg temperature difference, the flow per square foot of inlet or outlet is 165 cfm (Fig. 2), and $\frac{400{,}000}{165} = 2{,}400$ sq ft of opening required in the sidewall and an equal amount in the monitor. Say that only 2,000 sq ft can be readily accommodated in the two sides of the monitor. This is 83 per cent of 2,400. The inlets must then be 160 per cent of 2,000 or 3,200 sq ft (Fig. 4). With the building twice as long as it is wide, this quantity would be divided into about 1,000 sq ft for each side and 600 sq ft in each end.

Computations for Wind Only

Wind velocity = 8 mph (diagonal) = 704 fpm
Window openings in path of wind = 1,600 sq ft (one side and one end)
$0.35 \times 704 \times 1{,}600 = 394{,}240$ cfm

Computations for Combined Forces

When the wind is blowing, the windward side of the monitor should be closed, which gives the conditions of 1,000 sq ft in the upper openings, and 3,200 sq ft in the lower, a ratio of 3.2 to 1. The flow due to temperature difference must then be determined for that condition and the computation would be—

$1{,}000 \times 165 \times 1.35 = 222{,}750$ cfm (Fig. 3)
Adding the two flows:

Temperature difference	= 222,750 =	36 per cent
Wind	= 394,240 =	64 per cent
Total	616,990 =	100 per cent

The computed flow would then be about:

$1.9 \times 222{,}750 = 423{,}225$ cfm (Fig. 6)

The original flow due to temperature difference was 400,000 cfm with all windows open. The effect of the wind is to increase the flow to 423,225 cfm which, according to Fig. 1, reduces the temperature difference from the original 10 deg to about 9 deg, even though one half of the monitor windows is closed.

When little or no wind is blowing, all windows may be opened wide to utilize the full effect of temperature difference. However, when there is a side

or diagonal wind, the tendency is for air to enter on the windward side of the monitor and thus to decrease the amount that the leeward side can exhaust from below. By using judgment in the manipulation of the windows and closing upper inflow openings, the windows are utilized to the best advantage for effectively airating the building.

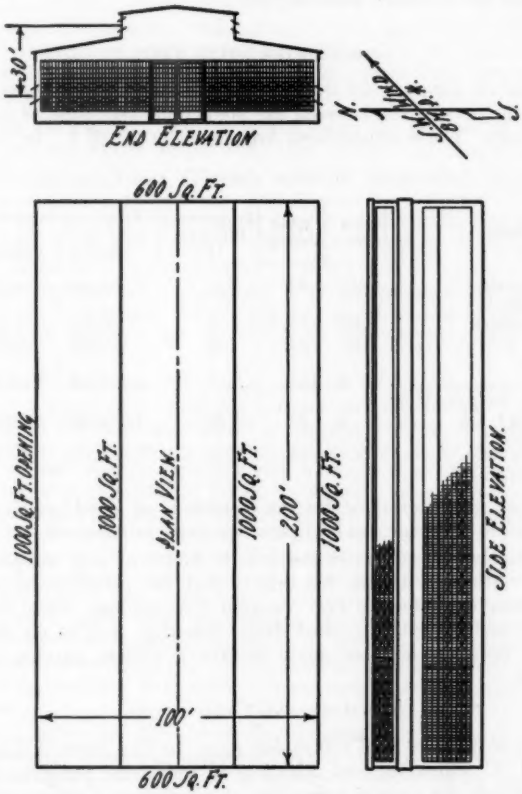

FIG. 7. TYPICAL INDUSTRIAL BUILDING

Some times the problem is one of effectively removing smoke, fumes or dust rather than of removing heat. This introduces an indeterminate factor impossible of mathematical solution unless experience has demonstrated just what air movement is necessary in the particular case in question. Even when considering the removal of heat, the airation is based on the judgment of what will be a satisfactory temperature difference. Predetermination methods, therefore, are predicated upon the selection of an air movement or temperature difference rather than upon definitely stated requirements.

It is occasionally found that the computed opening areas are small compared to those normally placed in a building. While they may provide the computed amount of air under the conditions for which the problem is solved, it is advisable to protect the design against possible change in use at a later date, and also to secure sufficient air for extremely hot days by providing wide openings through which the workmen also may see.

CHECKS WITH FIELD TESTS

The degree of accuracy of the predetermination methods given herein is indicated in the comparison between the predictions and some of the field tests that were made. These comparisons are shown in Table 1. In this table the

TABLE 1. COMPARISON BETWEEN OBSERVED AND COMPUTED VALUES

Building	Test No.	Wind Velocity	Temperature Difference (F)	Flow in Cu Ft per Min		Disagreement in Per Cent
				Observed	Computed	
U. of M. Foundry ..... Ann Arbor, Mich.	382	8.5	25	20,000	16,810	16
U. of M. Foundry	385	7.8	6	21,400	15,010	30
Riter-Conley Co. ..... Leetsdale, Pa.	261	3.2	0	335,000	293,715	12
D. S. P. Co. ..... Spring Plant, Detroit, Mich.	8	15.2	5	2,225,500	2,126,500	5
New Departure Co. ..... Bristol, Conn.	2	4.3	18	1,670,000	1,602,000	4

computed flows are determined for the conditions of wind velocity and direction, temperature difference and window openings as observed.

It is evident that the observed flow was in excess of that computed in all of these cases. The explanation for this is that the predetermination methods are based upon the effective flow through the building, while the observed flows usually include some so-called short circuiting, that is, air flowing in at certain high openings and out again at others without passing through the working zone.

SUMMARY AND CONCLUSIONS

This paper is intended as a companion paper to The Neutral Zone in Ventilation,[3] by J. E. Emswiler, and Airation of Industrial Buildings,[4] by W. C. Randall, in which the general principles of airation were discussed at length. Therefore, the authors have confined themselves to the presentation of a method for computing either the amount of window opening to install in a building to obtain a predetermined airation, or the airation to be expected from a given fenestration.

The size and shape of an industrial building are usually adapted to the desired production capacity and the nature of the industry. The fenestration should also be determined by these same items, if the natural forces are to be depended upon to produce the airation. The data in this paper make it possible to determine the window openings necessary to produce a selected air movement, or a desired temperature difference when the general building dimen-

sions and heat emission, as well as the velocity and relative direction of the prevailing winds are known. It also makes it possible to compare several designs for airation results and thus to select the most desirable one.

DISCUSSION

J. E. EMSWILER (WRITTEN): The subject of this paper is simply worked out and satisfactorily presented and it should be well received, because it shuns generalities and considers actual application. The basis on which the data are presented is sound and conservative.

R. J. LEAVER (WRITTEN): We are contemplating the erection of a building in which glass area now figures 53 per cent. The question has been raised as to the amount of heat which will be transmitted to the interior of the building from direct sun rays.

Have the authors any available data regarding the heat transmission through glass set in steel sash? Also have they any information relative to the proposed method of dissipating excess heat transmitted to the building?

A. VOGEL (WRITTEN): The authors are to be congratulated on maintaining the high standard set by preceding papers, and they are deserving of our appreciation in contributing data which we can use readily in improving the design of industrial buildings.

Experience in the designing of buildings using the information given in the paper has shown that the principles stated are a direct and valuable aid in design. The admission of air is the most difficult part in the application of the principles. It is absolutely necessary to arrange the openings for incoming air so that some of these openings can be used even in extremely cold weather. It is advisable in buildings where processes produce smoke or fumes, to arrange some of the openings at a sufficient height above the floor so that drafts will not affect workmen near the windows. Inasmuch as the temperature difference between inside and outside of a building is greater in the winter than in the summer, a minimum vertical distance between inlets and outlets is required in the winter and a maximum in the summer when it is necessary to have some inlets located as far down as possible. This lower row of inlets can be used freely during the summer because workmen near by have no objection to an incoming current of warm air. It appears, therefore, that in buildings such as foundries, forge shops, etc., 2 rows of inlets are desirable in the side walls, the lower to be used in warm weather and the upper to be used in cold weather.

Heating engineers should give considerable attention to the natural ventilation of industrial buildings obtainable by using the methods described by the authors. They should also consider the heat given off by a heating system, which rises to the top of an industrial building and produces a form of natural ventilation following the rules described in the paper. Heating engineers are thoroughly familiar with the fact that the temperature at the breathing line may be 65 F and be considerably higher at the top of the building. Of course, as this heated air is exhausted at the top of the building additional heat and fresh air must be added, but as heated air rises anyway it may as well be used for ventilation purposes with beneficial results. In using this means of ventilation, however, one of the principles stated in the paper, that the inflow must be equal to the outflow, must be observed, or undesirable and uncontrollable drafts will result.

If the principles are carefully observed in all phases of natural ventilation, not only in those phases produced by wind and by heat given off either by manufacturing processes or heating systems, but also by recirculation caused by the sun and wind

effects on windows and walls, better control of air currents in a large building by proper manipulation of the heating system will be obtained.

Heating systems in buildings of large volume must take into account natural ventilation consisting of inflow and outflow and also recirculation. A matter of some importance, which affects the design of buildings with sawtooth skylights, is that of the radiation located just below the glass surfaces to keep the waterways free from ice and the glass areas free from snow. This radiation affects natural ventilation by concentrating heat in such a place that warm air rises and is vented through the skylights. This heat loss, while appreciable, is actually beneficial because it promotes desirable ventilation. It is apparent, therefore, that natural ventilation is an important factor in various phases of the design of heating systems as well as the ventilation of large industrial buildings.

The paper already has demonstrated its importance with respect to natural ventilation. It is well, however, to add that heating and ventilating engineers are the men properly trained to assist architects and owners in the fenestration of buildings so that the principles of natural ventilation are taken advantage of and used to obtain the best possible results.

Illuminating engineers are giving attention to natural illumination as a basis of the solution of problems in artificial illumination. In the field of natural illumination Mr. Randall also has contributed excellent papers. It is logical, therefore, for heating and ventilating engineers to include natural ventilation as a part of their practice, and to use its principles in the design of the heating systems and mechanical ventilation of buildings.

L. A. HARDING: Mr. Chairman, personally this paper is of great interest to me and all of Mr. Randall's papers are of interest to any one doing industrial work or having anything to do with industrial design. He always sets a standard of presentation which I think could be profitably followed by other authors. When he includes formulae, he usually gives an example as to how the formula should be worked, with corroborative data on some actual installation. I think the paper is an excellent one.

W. C. RANDALL: The questions which Mr. Leaver asks are not within the scope of the intent of this paper. Frankly, we have no available data on the heat transmission through glass, particularly of the character which would allow you to figure on a method of dissipating the excess heat transmitted into a building in the summer time in order to maintain temperature conditions.

The data which we have available on heat transmission through glass was obtained in a comparison between different types of glass as to their heat transmitting and heat reflecting qualities, but this information is not adequate for the use which you have in mind.

Practically speaking, if the heat coming into the building through the glass is excessive, I would recommend that the glass be painted with a product which will not be washed off with the rain but which can be washed off in the fall with warm water.

In this paper you will notice the authors are sticking to this word *airation* as being not entirely synonymous with ventilation. In some cases it is possible that it is synonymous. We are sidestepping whether or not ventilation is accomplished and sticking to the text, the movement of air in, through and out of buildings.

No. 911

THE MEASUREMENT OF THE FLOW OF AIR THROUGH REGISTERS AND GRILLES

PART II

By L. E. DAVIES,[1] CHICAGO, ILL.
NON-MEMBER

This paper is the result of research conducted at Armour Institute of Technology of Chicago, in cooperation with the A. S. H. V. E. Research Laboratory and the Ventilating Contractors Employers Association of Chicago.

THE initial report covering the research work conducted in the Laboratories of the Armour Institute of Technology to determine the most satisfactory method of measuring the air passing through a register or grille, was presented at the 36th Annual Meeting of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, Philadelphia, Pa., January, 1930.[2] It appeared from this investigation that the only feasible device to be used for this purpose was the anemometer, and the problem therefore resolved itself into a study of that instrument.

For purposes of the investigation a centrifugal fan and a system of ducts were arranged in such a manner that the same air which passed through the grille also passed through a length of straight round duct in which Pitot tube traverses were made simultaneously with anemometer traverses at the face of the grille. Tests were conducted with the ducts and grilles both on the exhaust and discharge sides of the fan.

The tests previously reported were made entirely with one type of anemometer and were confined generally to thin lattice work grilles with various percentages of free area, although a few observations were taken with expanded metal lath, and with thick iron having circular openings.

The following conditions which were not considered in the original study were included in the investigation covered by the present report:

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[1] Associate Professor of Experimental Engineering, Armour Institute of Technology.

[2] Measurement of the Flow of Air Through Registers and Grilles, L. E. Davies, A. S. H. V. E. TRANSACTIONS, Vol. 36, 1930.

Presented at the Semi-annual Meeting of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, Swampscott, Mass., June, 1931.

1. To study some of the more ornate forms of grilles, particularly the ornamental plaster type.
2. To use the various types of anemometers on the market to determine whether or not the formulae previously derived could be applied equally well to all types.
3. To make tests at higher velocities than those previously attained.
4. To have traverses made by various individuals, particularly by those previously unfamiliar with the instrument, to determine the importance of the personal element.

In the earlier work it was found that with supply grilles the following formula could be used:

$$\text{cfm} = \frac{CV(A+a)}{2} \qquad (1)$$

in which

cfm = volume of air in cubic feet per minute
V = average indicated velocity obtained by anemometer traverse taken in contact with grille (after applying anemometer correction)
A = gross area of grille—square feet
a = net free area of grills—square feet
C = coefficient determined by experiment

The value of C was 0.952 at 150 ft per minute and approached 1.00 as the velocity increased. Thus, for most of the practical work a value of 1.00 is sufficiently accurate which makes the formula a very convenient one.

On exhaust work the results were not quite as satisfactory as somewhat different values of C were obtained with the different grilles. However, by means of an average value of C which varied with the velocity it was possible to obtain quite accurate results.

The following alternate formula, which was applicable only in the case of exhaust grilles, was also submitted:

$$\text{cfm} = KVA \qquad (2)$$

in which

V = average indicated velocity obtained by the anemometer traverse in contact with grille
A = gross area of grills—square feet
K = coefficient determined by experiment

The average value of K in formula (2) was 0.8, while for more accurate results a value ranging from 0.762 at 150 ft per minute to 0.832 at 700 ft per minute was recommended, with the value apparently approaching a constant figure at the higher velocities. At the time these two formulae were submitted there appeared to be little choice between them as to accuracy, neither one of them giving results as consistently accurate as those obtained with supply grilles, but both of them a great deal better than previous methods of calculation.

Figs. 1 to 4 show the types of grilles covered by this report. Fig. 1 shows a grille having the same face design and percentage of free area as one originally tested, but differing in that it was of cast iron several times thicker

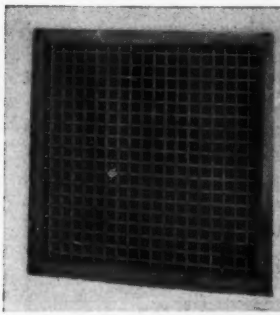

FIG. 1. CAST IRON GRILLE HAVING FREE AREA OF 60.5 PER CENT

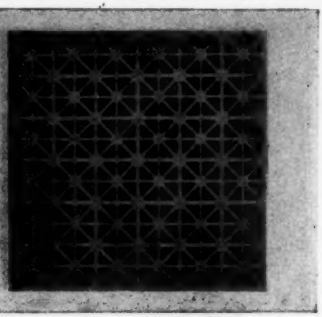

FIG. 2. THIN IRON GRILLE OF CONVENTIONAL ORNAMENTAL DESIGN HAVING FREE AREA OF 54.8 PER CENT

FIG. 3. PLASTER GRILLE HAVING FREE AREA OF 43.0 PER CENT

FIG. 4. PLASTER GRILLE HAVING FREE AREA OF 34.2 PER CENT

than the former one and having the usual tapered frets characteristic of castings. The gross area also differed somewhat from any grills previously used.

Fig. 2 shows a familiar type of thin iron grille having a conventional ornamental design. Figs 3 and 4 show two plaster grilles of quite different design. Fig. 3 is characterized by the fact that the large center portion has an extremely small amount of free area, while the four corners have open spaces as large as the anemometer. Certain portions of the pattern also were raised above the general surface level, so that it was impossible to keep the anemometer in intimate contact with the face of the grille at all times. The thickness of this grille was approximately the same as that of the cast-iron grille shown in Fig. 1. Fig. 4 shows a design which is distinguished by the largeness of the details. It is very thick, has rounded edges, and has both large free spaces and large solid portions.

TABLE 1. SUMMARY OF RESULTS WITH EXHAUST GRILLES WHEN USING FORMULA

$$\text{cfm} = CV\left(\frac{A+a}{2}\right)$$

Grille Used	Indicated Velocity fpm	C (From Curves)	Percent Error When Using $C=1.02$	Percent Error When Using C from Previously Recommended Values	Percent Error Using Formula cmf$=Va$
1. Cast Iron	150	0.997	+2.2	−0.3	−24.5
	400	1.012	+0.6	+3.5	−25.5
	700	1.021	−0.1	+6.0	−26.2
2. Ornamental Iron	150	0.998	+2.1	−0.4	−26.6
	400	1.060	−4.0	−1.0	−31.4
	700	1.097	−7.0	−1.0	−34.2
3. Ornamental Plaster ..	150	1.040	−2.0	−4.5	−40.5
	400	1.119	−8.8	−6.0	−44.4
	700	1.181	−13.8	−8.2	−47.0
4. Ornamental Plaster ..	150	1.075	−5.2	−7.5	−56.0
	400	1.152	−11.5	−9.0	−58.5
	700	1.211	−16.0	−10.5	−59.7

TABLE 2. SUMMARY OF RESULTS WITH EXHAUST GRILLES WHEN USING FORMULA

$$\text{cfm} = KVA$$

Grille Used	Indicated Velocity fpm	K (From Curves)	Percent Error When Using $K=0.8$	Percent Error When Using K from Previously Recommended Values	Percent Error Using Formula cmf$=Va$
1. Cast Iron	150	0.801	−0.1	−5.0	−24.5
	400	0.813	−15	−0.6	−25.5
	700	0.820	−2.5	+1.4	−26.2
2. Ornamental Iron	150	0.748	+7.0	+1.9	−26.6
	400	0.799	+0.1	+1.0	−31.4
	700	0.834	−4.0	−0.2	−34.2
3. Ornamental Plaster ..	150	0.724	+10.5	+5.0	−40.5
	400	0.773	+3.2	+4.1	−44.4
	700	0.814	−0.5	+2.1	−47.0
4. Ornamental Plaster ..	150	0.779	+2.9	−2.0	−56.0
	400	0.824	−2.8	−2.0	−58.5
	700	0.848	−5.6	−1.9	−59.7

EXHAUST GRILLES

The results obtained with the different types of grilles on exhaust are shown by the curves of Figs. 5 and 6. These curves were plotted from a large number of tests conducted in the same manner as described in the original report.[3]

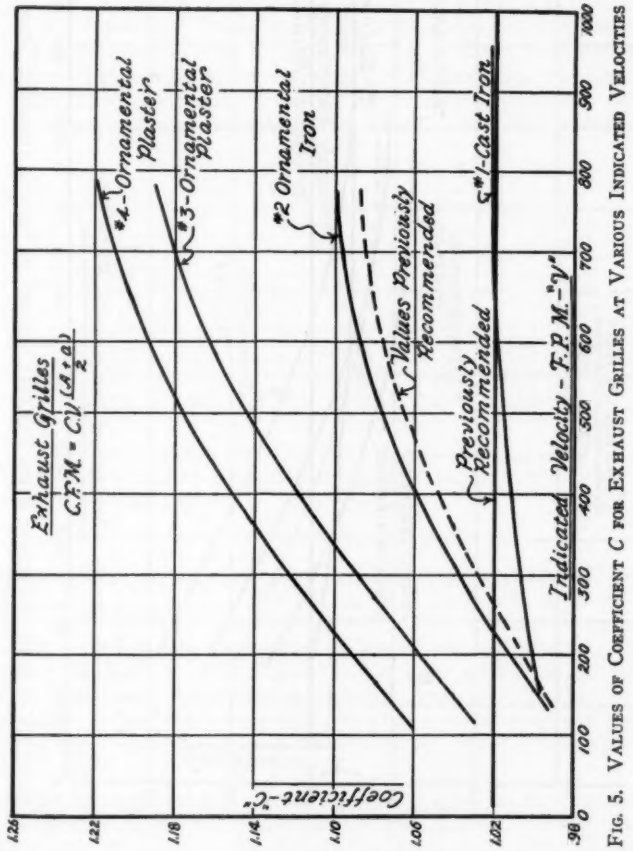

FIG. 5. VALUES OF COEFFICIENT C FOR EXHAUST GRILLES AT VARIOUS INDICATED VELOCITIES

In the case of the iron grilles, the individual values of C and K varied above and below the curves shown by about the same amount obtained in the earlier tests, but in the case of the plaster grilles this variation was somewhat greater

[3] Measurement of the Flow of Air Through Registers and Grilles, L. E. Davies, A. S. H. V. E. TRANSACTIONS, Vol. 36, 1930.

though not to a serious degree. The results are summarized in Tables 1 and 2. Table 1 shows the results obtained when using formula (1). It will be noted that the results obtained with the two iron grilles were in close agreement with the earlier results except that the curve for the cast-iron grille is much

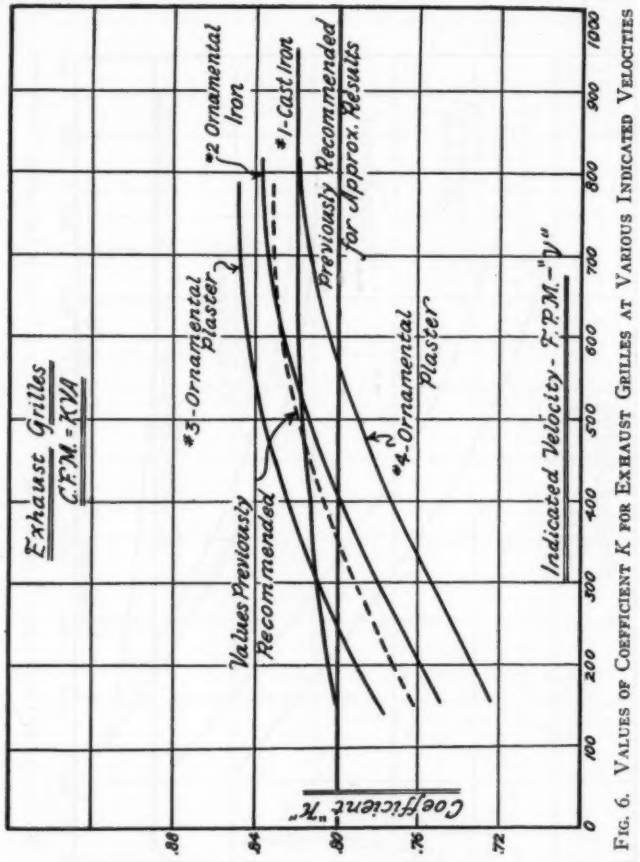

FIG. 6. VALUES OF COEFFICIENT K FOR EXHAUST GRILLES AT VARIOUS INDICATED VELOCITIES

flatter than any of the others. The results obtained with the plaster grilles were less satisfactory although these results were a great deal more accurate than those obtained by means of the earlier methods of calculation, as can be seen by a comparison with the figures in the last column.

The percentage of error when using formula (2) is considerably less as shown by Table 2. So far as the iron grilles are concerned there is little choice

between the two formulae, but with the plaster grilles the results obtained with formula (2) are much more accurate.

Considering all of the work completed up to the present time, formula (1), when used with the recommended values of C for the various velocities, gives maximum errors of ±11 per cent while formula (2) with recommended values of K reduces this error to ±5 per cent. When using constant values of 1.02 and 0.8 respectively, formula (1) will be subject to errors of from +13.5 per cent to —15.5 per cent, while formula (2) will give results varying from +11 per cent to —7 per cent. From these figures it is apparent that formula (2) is to be preferred for exhaust measurements. It is rather fortunate that this is the case as a knowledge of the percentage of free area is not required. This latter value is extremely difficult to measure or estimate with any degree of accuracy when the grilles are elaborate in design, so that any formula involv-

TABLE 3. TESTS ON ORNAMENTAL PLASTER SUPPLY GRILLES

Grille No.	Indicated Velocity fpm	Pitot Tubes cfm	Anemometer Traverses						C
			Va		VA		$\left(V\frac{A+a}{2}\right)$		
			cfm	Percent Error	cfm	Percent Error	cfm	Percent Error	
3	349.5	802	578	—28	1,344	+ 68	961	+20	0.834
4	300	556	411	—26	1,200	+116	805	+45	0.690
	566	1,130	775	—31	2,264	+100	1,519	+34	0.744
	891	1,785	1,222	—32	3,564	+100	2,393	+34	0.746

ing the free area would necessarily be subject to the errors made in the measurement of that value.

SUPPLY GRILLES

It is unnecessary to show curves or elaborate data covering the work done on the two iron grilles when used for supply. The results coincided exactly with the results obtained in the earlier tests. The results obtained with the ornamental plaster grilles when tested on the supply end, however, were less satisfactory inasmuch as the values of C for these grilles were considerably below 1.0 in the recent tests, whereas in every other case they were very close to this value. It has not been possible to give a satisfactory explanation of this fact. The results of a few of the observations made on these grilles are shown in Table 3.

It will be noted that with the grille shown in Fig. 4, the first condition was encountered under which the results obtained with formula (1) were less accurate than with the earlier method, namely that of multiplying the indicated velocity by the net free area. Many runs were made with variations in the method of taking the traverse and with the different types of anemometers, but with substantially the same results. The only consolation in these unsatisfactory results lies in the fact already discussed, namely, that the difficulty in determining the amount of free area with this type of grille would make the results more or less inaccurate even though the method were inherently correct.

It appears both desirable and necessary to device some better method of handling this type of grille. The author is engaged at the present time in the development of such a method and has recently obtained encouraging results.

Some time has been devoted to the study of higher velocities ranging upward to 1600 ft per minute. As anticipated, the increases in the values of C were

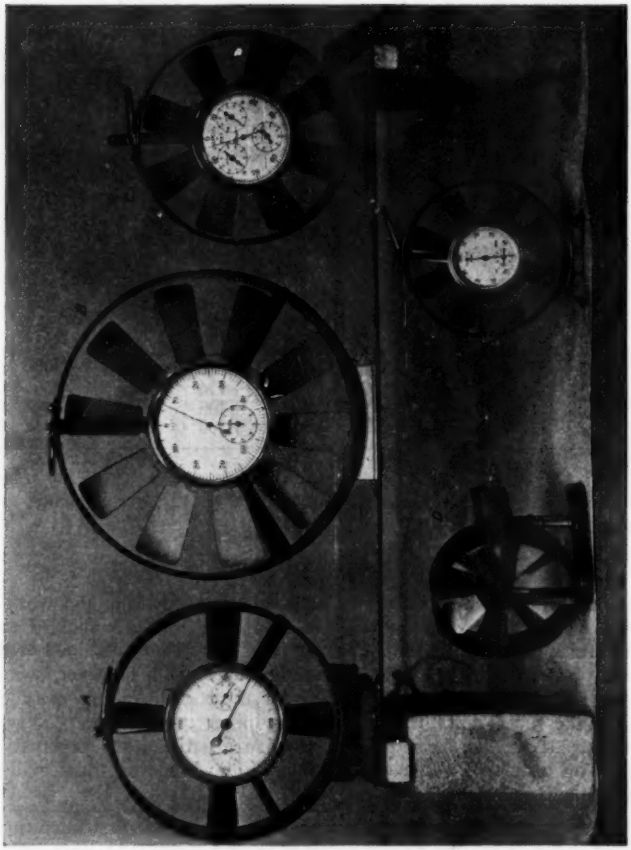

FIG. 7. TYPES OF INSTRUMENTS TESTED

very slight. This fact was recently verified in connection with a unit heater test. The heater was of the draw-through type and discharged the air through a nozzle-like outlet 24 x 6 in., in which were mounted numerous adjustable horizontal and vertical baffles. These baffles were flat and when adjusted properly gave an extremely uniform air flow, with a velocity of about 2200 ft per minute. The straight uniform flow, together with the high velocity, made

it possible to take Pitot tube readings at the outlet. The volume of air per minute as calculated from these readings checked perfectly with the value obtained from an anemometer traverse when using the factor 1.0 in the formula. Furthermore, a very good heat balance served to verify the accuracy of the measurements.

COMPARISON OF ANEMOMETERS

All of the work discussed so far has been executed with the 4-in. American made instrument of Type *c* in Fig. 7. During the course of the work three of these instruments were used. Through the courtesy of the manufacturers, one of each type handled by them was obtained. Table 4 identifies the various instruments used.

Direct comparisons were made between the various instruments, that is, the

TABLE 4. COMPARISON OF ANEMOMETERS

Type	Face Diameter, Inches	Rated Velocities Feet per Minute	Relative Values of C or K
a	4.37	500 to 6,000	1.00
b	6.00	75 to 500	1.04
c	4.00	200 to 3,000	1.00
d	2.75	200 to 3,000	0.98
e	3.00	200 to 1,000	0.98

traverses with the instruments being compared were made in rapid succession without any change in the conditions. The results obtained were rather interesting in that it appears that the coefficients are, to a small degree, a function of the face diameter. Under identical conditions the two smallest instruments invariably agreed very closely in their indicated velocities, with the smallest one tending to show the largest reading. The difference, however, was very slight in spite of the radical difference in their design. Similarly type *c* and type *a*, the high velocity instruments, checked each other perfectly, but with all readings averaging approximately 2 per cent lower than the smaller instruments. The greatest discrepancy occurred between the 4 in. and the 6 in. low velocity types. The latter indicated velocities averaging about 4 per cent lower than the former. The last column of Table 4 shows the effect of these variations on the value of the coefficient. Since the normal value of C is less than 1.00 at velocities within the range of the 6-in. anemometer, it is suggested that a coefficient of 1.00 always be used with this type of instrument, in which case the errors will be compensating ones, and the resulting net error reduced to a small value.

As a test of the importance of the personal element, during the past few months 17 groups of students, four men in each group, repeated some of the work, using various grilles, anemometers, and both exhaust and supply ducts. Each group secured two sets of data. In each case, careful instructions were given as to exactly what was to be done, and the Pitot tubes and draft gages were properly adjusted and the students then left to their own resources. In spite of the fact that most of these men had never handled an anemometer before, there were only three results out of the total of 34 that were farther from the average curves than the author's tests had been. This indicates that

when a definite method of procedure is outlined and followed the personal element is largely eliminated. Unfortunately, many times the worker is impatient and desires to secure results with a minimum expenditure of time and effort. In such cases he will take a very hurried moving traverse with the result that his figures are unreliable. As previously stated, a moving traverse may be made accurately providing enough time is allowed and a uniform speed is maintained.

It is when unusual conditions are encountered and the testing engineer must depend upon his own judgment that the personal element becomes of greatest importance. It is safe to say that no set rules or formulae can be devised that can be used with absolute accuracy under all conditions. The good judgment of an intelligent man is an essential part of every test whether it be in air measurements or in any other branch of engineering.

DISCUSSION

E. H. BELING (WRITTEN): The author has made a most favorable contribution to the industry. His method for measuring the flow of air through exhaust grilles can be used easily in the field and it is accurate enough for all ordinary work. We are pleased to note that by using the formula $\text{cfm} = KVA$ it is not necessary to determine the net free area. As the author states, this always has been a difficult measurement and its determination probably has caused as many inaccuracies as any other factor.

The author has covered the field of exhaust grilles when he used the 4 types illustrated in Figs 1 to 4. It seems logical to assume that any other ordinary grille would have similar characteristics.

In continuing his investigation on the measurement of air flow through various types of supply grilles, we hope the author is successful in arriving at as simple a method as he has found for exhaust grilles, and that his investigations will include deep nozzle type supply grilles such as are used on high velocity unit ventilators. This information will be necessary before any great progress can be made in the matter of field testing of unit ventilators.

There also is some work to be done on the subject of measuring the air discharged by unit heaters by the use of anemometers. At present it is practically impossible to make even an intelligent guess as to the air delivery of unit heaters without a laboratory set-up.

Concerning the author's comparison of results obtained when using the different anemometers illustrated in Fig. 7, we are pleased to find that for practical purposes they deliver identical results. Many opinions have been expressed as to the relative merits of the different anemometer sizes and we are glad this question has been settled.

We are agreeably surprised with the author's findings in his work on groups of students using anemometers. If the present element was of great importance, the anemometer method of air measurement would probably pass into discard.

S. I. ROTTMAYER (WRITTEN): The coefficient C and K for plain lattice cast-iron grilles varies but little over a wide range of indicated face velocities. This style of grille is by far the most commonly used in ventilating work. For ornamental iron and ornamental plaster grilles, however, the values of the coefficients bear a relationship to indicated face velocities which is quite different from that relationship for plain lattice grilles. No reasonable explanation for this phenomenon is apparent. The magnitude of this coefficient seems to be affected largely by the physical design of the ornamental face.

The common range of face velocities over grilles used in ventilating work may be said to be from 250 fpm to 600 fpm. Much practical value might ensue if the manufacturers of grilles, in cataloging their product, would publish a curve of the coefficient values over the practical range of indicated face velocities, or give two or three values within this range, for each design of grille which they sell. Anemometers are sold with such information in the form of calibration curves.

Results then might be expected for air quantity measurement for a given grille that would be within the range of accuracy of the anemometer.

A. L. WEIXEL (WRITTEN): The results of Professor Davies' experiments as submitted in this paper, will be received with satisfaction by the entire ventilating industry.

The necessity for accurate data on this subject has long been recognized, but it was not until Professor Davies submitted his original paper at Philadelphia that members realized the margin of error prevalent in the various methods in use for measuring air flow through registers and grilles.

Since the adoption in Chicago, of Professor Davies' original formula *cfm equals velocity, times gross plus net over two,* by the *Ventilating Contractors Employers Association* and the Ventilation Division of the Chicago Department of Health, for plain lattice supply openings, a great deal less friction has existed between contractors and city inspectors. The balancing of systems has been simplified, and as near as possible, anemometer readings have been standardized when making the final test.

Considerable argument has resulted in regard to the accuracy of this formula when used on exhaust systems, or when used on supply or exhaust systems having ornamental iron or plaster grilles.

Happily, the author answers this question with an accurate and practical method of introducing a constant, based on the percentage of free area in an ornamental grille on a supply system, and the extremely simple formula for exhaust ornamental grilles of, *eight tenths of the gross area, times velocity, equals cfm* which is sufficiently accurate for all practical purposes.

Contractors testing or balancing a ventilation installation are not interested in laboratory apparatus or methods that will give accurate results within 4 decimals. They want a readily understandable, practical method, reasonably accurate, that can be used on the job by the mechanics who install the job, without the contractor wondering whether they understand what they are performing, or whether his installation when so tested by his own men would *get by the city.*

In Chicago, where a variation of 5 per cent above or below specified requirements is permitted by the city's *Division of Ventilation,* I recently tested several large ventilating installations and found no difficulty in training a large group of contractors' employees, most of whom have only a common school education, to use the *gross plus net over two* formula, and secure readings that rarely vary more than 3 or 4 per cent from those secured by the city's inspectors, when testing supply openings with plain lattice grilles.

The widespread adoption of the constants recommended for ornamental iron and plaster grilles on both supply and exhaust will do much to eliminate the disputes that now arise from mistaken methods of establishing the basis of calculations.

The writer believes the Society should exert its influence with register and grille manufacturers to have them publish in their catalogs the formulae recommended by Professor Davies, together with typical examples of computing the cfm obtained at the varying velocities. Fan manufacturers should also be invited to include this information in their catalogs. This is probably the easiest way to secure the immediate general adoption of the results of Professor Davies' experiments which have

satisfactorily solved a problem that has given a great deal of trouble to the entire ventilating profession.

E. P. WELLS (WRITTEN): It is good news to learn that Mr. Davies is planning to continue his valuable investigations into the use of the anemometer. The difficulties that are encountered in measuring air volumes in various parts of air conditioning systems are such as to require an intimate knowledge of the conditions required to get accurate readings.

My experience with an attempt to measure the amount of air passing through a diamond-shaped outlet having holes and bars 3 in. to 6 in. wide has shown me that passing an anemometer across the face of such a grille gives meaningless results. Often the duct work is not accessible for pitot tube readings. In cases of this sort I have used a short piece of ductwork extended beyond the grille, of a length equal to at least the narrowest width of the grille. About a half of the length of the duct, next to the grille, was filled with vanes about 2 in. apart, to give parallel flow to the air. The full area of the outlet of the duct was used in figuring the volume. I have never checked the results with any accurate method of measuring.

Another problem that deserves solution is: How near to the side of an open duct will an anemometer read accurately? Imagine a circular duct 6 ft in diameter, with an anemometer placed in the center. We know that the instrument will read the velocity at that point correctly. We also know that the velocity of the air passing through the blades is not as great as that of the air passing by, because of the resistance offered by the blades. If, now, we keep the velocity constant, and reduce the diameter of the duct until it equals the diameter of the anemometer, all the air will finally pass through the anemometer and it is apparent that the instrument will record a higher velocity than actual. What is the smallest diameter of duct which will give a true reading? In measuring the air through a 4-ft round duct, placing the anemometer at the end, I found that a 3-in. anemometer gave a reading ¾ as great as a 4-in. anemometer.

L. E. DAVIES (WRITTEN): Regarding the diamond shaped outlet referred to by Mr. Wells I imagine that the large openings and large bars would place it in much the same class with the large plaster grille with which I obtained excessively large errors when using my formula. If Mr. Wells will send me a simple sketch showing more in detail the construction of this grille, I shall attempt to investigate it in the near future. We realize, of course, that there are many special cases where any of the methods or formulae discussed so far will be inaccurate and it is my intention in my next paper to classify and discuss in detail these special problems. In some cases only a modification in the procedure is needed, while in others a distinctly new method will be necessary. I have been experimenting with an attachment to be used in conjunction with the anemometer which has given promising results so far, but I wish to conduct further tests with it before offering it to the Society.

Regarding Mr. Wells's use of a duct at the outlet of a grille, there are several possible sources of error in this method. In some cases the presence of this duct will increase the flow through the grille. Whether or not this will be large enough to be serious will depend largely upon other characteristics of the system (fan, ducts, approach, etc.). Usually I have found in such cases a narrow border of low velocity or re-entering air which the anemometer does not reach effectively, tending to make the calculated flow too large. Furthermore, the effect of the grille, I have found, extends out for a surprisingly great distance.

I have recently used this method in investigating unit ventilators, using a duct length about 5 times that of the smallest grille dimension. When this duct was made straight with the outlet equal to the gross area of the grille, and the anemometer traverse multiplied by gross area, the resulting figure was 20 to 25 per cent too

high. When the duct was tapered to an outlet area approximately equal to the free area of the grille the results were quite satisfactory.

Mr. Wells's question as to how close to the side of an open duct an anemometer will read correctly can only be answered indirectly. When the air velocity at various points across the face of an anemometer varies considerably, the indicated velocity will be approximately midway between the maximum and the true average velocity. The accuracy of a measurement taken at the side of a duct will depend largely upon the rate at which the velocity varies within the space covered by the instrument. The presence of the duct wall itself will have little direct influence. However, where the instrument is located in a small circular duct, the effect of the walls completely surrounding the instrument becomes appreciable, resulting in an indicated velocity approximately 20 per cent too large when the duct reaches a diameter equal to that of the anemometer. E. Ower in his book, Measurement of Air Flow, recommends that the anemometer should not be used in a duct smaller than 6 times the instrument diameter. This may be a fair figure where the instrument is wholly within the duct, but where the measurements are made right at the outlet, I know from experience that accurate measurements may be made with ducts not over 3 or 4 diameters. On two occasions recently I have been able to verify this fact.

Mr. Wells states that he has found that a 3-in. instrument placed at the end of a 4-ft duct gave a reading only ¾ as great as a 4-in. instrument. There must have been some special conditions present to account for this, as it is not true in general. I have a 14-in. duct in my laboratory which I have used regularly for checking one instrument against another and have never noted such a discrepancy. I have just received two instruments (one of each type) from the factory where they were adjusted and newly calibrated and have this morning compared the two at the outlet of this 14-in. duct, both by spot readings at the center and by traverses from wall to wall. In spite of a rather non-uniform velocity distribution, the two instruments differed by only 2 per cent in both cases, with the 3-in. instrument showing the largest reading.

Regarding the deep nozzle type of unit ventilator with high outlet velocity mentioned by Mr. Beling, I can say with assurance based on actual experience with one, that the usual method and formula $\frac{V(A+a)}{2} = \text{cfm}$ may be used with accurate results. It is with the lower velocity types such as the *Univent* and the others of similar character that greater difficulties are encountered due to the close proximity of the heater core to the grille. My experience with these units will, I believe, be discussed by John Howatt in his report on the unit ventilator code.

We are working on a device or attachment to be used in conjunction with the anemometer in cases where the normal procedure is not applicable for any reason, as for example in cases where it is impossible to determine the free area accurately. One of the applications for this device which I have had in mind is the direct test of unit heaters. Although the results so far are encouraging, I do not care to make any positive claims until further tests have been run.

In Memoriam

NAMES	JOINED THE SOCIETY	DIED
EDWARD ADDY	1923	Dec. 1931
R. CARTER BEVERLEY	1905	Mar. 1931
WALLACE M. COSGROVE	1923	June 1931
DWIGHT H. FAULKNER	1926	Feb. 1931
JOHN H. GASSLER	1927	Nov. 1931
FRANK HUCKEL, JR.	1920	Oct. 1931
BURT MACKENZIE	1924	Jan. 1931
PAUL E. MUELLER	1919	Nov. 1931
JOHN MUSAUS, JR.	1923	Aug. 1931
C. EDWARD PEARCE	1911	Sept. 1931
ROBERT W. PRYOR, JR.	1913	July 1931
FRED R. RIEGER	1930	Dec. 1931
JOHN R. WEAVER	1930	June 1931
HERBERT W. WHITTEN	1908	Jan. 1931
JOHN J. WILSON	Charter Member	Dec. 1931

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